AD-A132 175 THE EFFECT OF CONDENSATE INUNDATION ON STEAM 1/2
CONDENSATION HEAT TRANSFER TO WIRE-WRAPPED TUBING(U)
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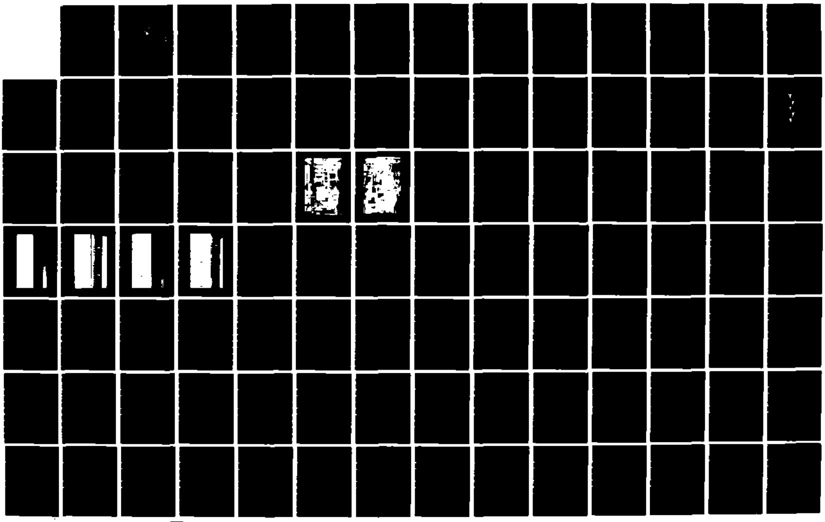

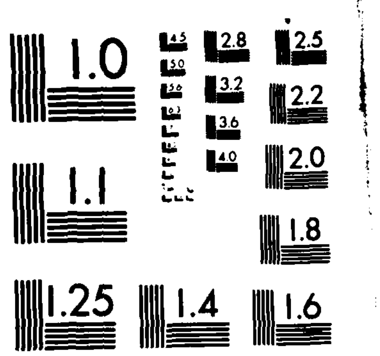

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THESIS

THE EFFECT OF CONDENSATE INUNDATION ON
STEAM CONDENSATION HEAT TRANSFER
TO WIRE-WRAPPED TUBING

by

Georgios Dimitriou Kanakis

June 1983

Thesis Advisor: P.J. Marto

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER	2. GOVT ACCESSION NO. AD-A132 175	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) The Effect of Condensate Inundation on Steam Condensation Heat Transfer to Wire-Wrapped Tubing		5. TYPE OF REPORT & PERIOD COVERED Master's Thesis June 1983
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Georgios Dimitriou Kanakis		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		12. REPORT DATE June 1983
		13. NUMBER OF PAGES 160
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report)
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Smooth, roped, wire wrapped tubes, inundation.

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

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Block 20 (continued)

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The Effect of Condensate Inundation on Steam Condensation Heat Transfer to Wire-Wrapped Tubing

by

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Submitted in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
June 1983

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Dean of Science and Engineering

ABSTRACT

Steam condensation heat transfer measurements were made in a 5-tube test condenser having an additional perforated tube to simulate up to 30 active tubes. Results were obtained for smooth tubes and roped tubes wrapped with wire. A Sieder-Tate equation was used to correlate the inside heat-transfer coefficient. For smooth tubes, a leading coefficient of 0.029 was found, while it was 0.061 for the roped tubes. The average condensing coefficient measured for 30 smooth tubes was 0.59 times the Nusselt coefficient calculated for the first tube. When the smooth tubes were wrapped with wire, this ratio increased up to 0.86. Further, roped tubes without wire experienced a ratio of 0.63, while roped tubes wrapped with wire resulted in a ratio of 0.86. These preliminary data show that wire-wrapped tubes may lead to a significant reduction in condenser surface area.

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NOMENCLATURE

A_o	Outside, heat-transfer area of one tube (m^2)
A_i	Inside, heat-transfer area of one tube (m^2)
C_i	Sieder-Tate coefficient
C_{pw}	Specific heat of water evaluated at T_b (KJ/kg·K)
C_f	Correction factor $(\mu/\mu_w)^{0.14}$
D_i	Inner diameter of the tube (m)
D_o	Outer diameter of the tube (m)
g	Acceleration of gravity (9.81 m/s^2)
h_i	Experimentally-determined value for the inside, heat-transfer coefficient ($W/m^2 \cdot K$)
h_{fg}	Latent heat of vaporization (KJ/kg)
h_N	Local, outside, heat-transfer coefficient for the Nth tube (W/m^2K)
h_{Nu}	Heat-transfer coefficient calculated from the Nusselt equation ($W/m^2 \cdot K$)
h_1	Outside heat-transfer coefficient for the first tube ($W/m^2 \cdot K$)
k_f	Thermal conductivity of the condensate film (W/m·K)
k	Thermal condictivity of the cooling water evaluated at T_b (W/m·K)
k_m	Thermal conductivity of titanium (W/m·K)
L	Condensing length (m)
LMTD	Logarithmic Mean Temperature Difference (°C)
m	Slope of the least-squares-fit, straight line
$\dot{m}$	Mass flow rate of cooling water (kg/min)

N	The number of tubes in a column or the tube number of a given tube
Nu	Water-side Nusselt number
P_r	Prandtl number evaluated at T_b
Q	Heat transfer rate (W)
q"	Heat flux based on outside area (W/m^2)
R_e	Water-side Reynolds number
R_f	Fouling thermal resistance (m^2K/W)
R_w	Wall thermal resistance based on the outside area (m^2K/W)
R_1	Outside, local, heat-tranfer coefficient ratio (h_N/h_1)
R_2	Outside, average, heat-tranfer coefficient ratio ($\bar{h}_N/h_1$)
S/D	Spacing-to-diameter ratio of tubes
T_b	Average cooling water bulk temperature (°C)
T_{ci}	Cooling water inlet temperature (°C)
T_{co}	Cooling water outlet temperature (°C)
T_w	Wall temperature (°C)
T_f	Average condensate film temperature (°C)
T_{sat}	Saturation temperature of steam (°C)
T_v	Vapor (steam) temperature (°C)
U_n	Overall heat-transfer coefficient (m^2K/W)
U_o	Outside heat-transfer coefficient (m^2K/W)
V_w	Cooling water velocity (m/s)
X	Sieder-Tate parameter ($X = Re^{+0.8}Pr^{+1/3})(\frac{\mu}{\mu_w})^{0.14}$
ΔT	Temperature difference (T_w-T_b) (°C)

Greek Symbols

ζ	Heat capacity parameter ($C_p \Delta T/h_{fg}$)
μ	Dynamic viscosity of water evaluated at T_b ($N \cdot s/m^2$)
μ_f	Dynamic viscosity of condensate evaluated at T_f ($N \cdot s/m^2$)
μ_w	Dynamic viscosity of water evaluated at T_w ($N \cdot s/m^2$)
ξ	Acceleration parameter ($K\Delta T/\mu h_{fg}$)
ρ	Density of cooling water (kg/m^3)
ρ_v	Vapor density (kg/m^3)

ACKNOWLEDGEMENT

Any work of this sort necessarily represents the influence of many people. I am especially indebted to Professor Paul J. Marto, my Thesis Advisor, for his support and patient guidance throughout the project.

My sincere appreciation to Dr. A.S. Wanniarachchi, my Second Reader, for his great help in overcoming even the smallest obstacles.

Special thanks to Mr. Thomas Christian, Mr. John Moulton, Mr. Willard Dames, and Mr. Ron Lonqueira for their technical assistance.

I wish to thank my wife Vassia and my family for their moral support, understanding and encouragement. Without it, this project certainly would not be complete today.

I. HISTORICAL BACKGROUND

In recent years, there has been a continued interest in the reduction of the size and weight of propulsion systems aboard both surface vessels and submarines. Especially, actual dimensions of naval condensers have a critical bearing on cost and performance of the ship. Often, compactness is more important than thermal effectiveness when the overall performance of the ship is considered; in a submarine, the diameter of the pressure hull can depend on the dimensions of the condenser.

The importance of compactness justifies measures to raise the overall heat-transfer coefficient of condenser tubes despite the penalties which may occur, i.e., the increased pumping power and tube cost.

Naval condenser design is based upon the Heat Exchange Institute (HEI) specifications for steam condensers [Ref. 1] and also the standards of the Tubular Exchange Manufacturers Association (TEMA) [Ref. 2]. Search [Ref. 3] investigated the present condenser design processes, including the feasibility of enhanced heat transfer in naval condensers. He concluded that the current design is very conservative, and he predicted that a forty-percent reduction in condenser weight and volume could be achieved depending on the heat-transfer enhancement method used.

In recent years, many research efforts have been directed to the study of heat-transfer enhancement techniques and their application to heat-exchanger design. Webb [Ref. 4] has summarized extensive works of augmentation techniques. At the Naval Postgraduate School, Beck [Ref. 5], Pence [Ref. 6], Reilly [Ref. 7], Fenner [Ref. 8] and Ciftci [Ref. 9] conducted experimental research into various kinds of enhancement schemes employing a single-tube test condenser.

The above-mentioned investigations concluded that, for the same diameter tube, the overall heat-transfer coefficient of enhanced tubes can exceed those for smooth tubes by almost 100 percent. Reilly and Fenner [Ref. 10] revealed that most of the above mentioned augmentation occurred on the cooling-water side due to a combination of increased surface area, and increased turbulence and swirl in the cooling water flow. Little or no improvement occurred on the steam side. Eissenberg [Ref. 11] performed an extensive study on condenser-tube, heat-transfer coefficients using a multi-tube bundle.

In order to investigate the outside heat-transfer performance of various enhanced tubes in tube bundles, research was conducted at the Naval Postgraduate School. Noftz [Ref. 12] modified a test apparatus initially designed by Morrison [Ref. 13] to simulate a tube bundle using five active tubes arranged in a vertical plane. A perforated tube was located at the top of the bundle, through which water was flooded to simulate bundles having up to 30 tubes in a vertical row. His investigations determined that the heat-transfer coefficients for a given tube of the tube bundle increased as the mean vapor velocity increased, but decreased as the amount of condensate inundation increased. The experimentally found values for the heat-transfer coefficients were comparative with the Nusselt theory.

Based upon research done currently, it is evident that present day smooth-tube steam condensers, operating under typical conditions, have limitations in their thermal efficiency, due to a large thermal resistance which occurs on the tube side of the condenser. This resistance is generally larger than any of the thermal resistances that occur on the steam side, in the tube wall, those due to fouling, or due to noncondensable gases. However, employing enhanced tubes, the inside heat-transfer coefficient can be increased by 100 to 200 percent over the smooth-tube case. The outside heat-transfer coefficient, on the other hand, is increased by only 10 to 50 percent. In this situation, the thermal resistances on the inside and outside of the tube can be approximately equal.

Webb [Ref. 4] reported that the dominant thermal resistance in film condensation is that of conduction across the condensate film and, therefore, a surface geometry that promotes reduced film thickness will provide enhancement.

Thomas, et al, [Ref. 14] tested ammonia condensation on a smooth tube with a wire wrapped in a helical manner. The measured condensing coefficient was approximately three times that predicted by the Nusselt equation for a smooth tube. Surface-tension forces draw the condensate to the base of the wires, which act as condensate run off channels.

Webb [Ref. 4] stated that, when noncondensables are present, an additional thermal resistance is introduced in the gas at the vapor-liquid interface. Mixing in the gas film will substantially reduce this thermal resistance. Therefore, the maintenance of high vapor velocities, or special surface geometries that promote a higher heat-transfer coefficient in the gas film will substantially alleviate the performance deterioration due to noncondensables.

Cunningham [Ref. 15] presented in his paper that, for the roped tubes on the vapor (shell) side, the enhancement is achieved by improved condensate drainage, while on the coolant (tube) side the helical ridges increased turbulence and, as a result, the inside convective coefficient.

Improvements on the condensing side up to 100 percent have been reported for single-tube tests [Refs. 16;17]. Although titanium has a low thermal conductivity, it provides a high resistance to erosion and water-side fouling. Titanium tubes with enhancement both inside and outside are commercially available through the Wolverine Tube Division of Universal Oil Products, Inc. The applications of these tubes having all the inherited properties of titanium are promising for naval condensers.

The goals of this thesis were therefore to:

1. Obtain baseline heat-transfer performance data for the test condenser utilizing 16 mm O.D. smooth titanium tubes.

2. Conduct steam condensation tests with the following enhanced tube geometries to determine steam-side heat-transfer coefficients in relation to smooth-tube performance:

    a. Wolverine "roped" tubes.

    b. Wolverine "roped" wrapped with titanium wire.

    c. Smooth tubes wrapped with titanium wire.

II. THEORETICAL BACKGROUND

The combined effect of vapor shear and inundation on the condensate film heat-transfer coefficient for cylindrical, horizontal tubes within tube bundles is a very complex and still insufficiently-understood subject, which is of importance to the efficient design of steam condensers. Although many researchers have studied this subject both theoretically and experimentally, there is no accurate methodology available for predicting the condensate-film, heat-transfer coefficient within tube bundles.

In 1916, Nusselt conducted his pioneering analysis for the simple case of condensation occurring on the outside of a single, isolated, horizontal tube. He idealized the problem by making the following assumptions for single tubes as stated by Nobbs [Ref. 18].

1. The wall temperature is constant.
2. The flow is laminar in the condensate film.
3. Heat transfer in the condensate is by conduction, and subcooling may be neglected.
4. The fluid properties are constant within the condensate film.
5. The forces due to hydrostatic pressure, surface tension, inertia, and vapor-liquid interfacial shear are negligible when compared to the viscous and gravitational forces.
6. The surrounding steam and vapor/liquid interface are at saturation temperature.

7. The film thickness is small when compared with normal tube diameters and the effects of curvature are small.

Based on the above assumptions, Nusselt predicted the famous relationship for the heat-transfer coefficient:

$$h_{Nu} = 0.725\left[\frac{k^3\rho(\rho-\rho_v)h_{fg}\cdot g}{\mu D\ (T_{sat}-T_w)}\right]^{1/4} \tag{2.1}$$

In order to simulate and analyze a tube bundle, Eissenberg [Ref. 11] stated the following additional assumptions:

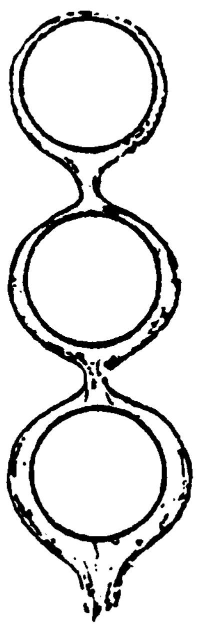
8. Condensate drains as a laminar sheet from a tube on to the tube directly underneath in such a way that velocity and temperature gradients are not lost in the fall between tubes.

9. The saturation temperature and the tube-wall temperature are constant for all tubes in the bank.

Jakob [Ref. 19] extended the Nusselt analysis for filmwise condensation heat transfer on a vertical in-line row of horizontal tubes as shown in Figure 1a.

The above-mentioned assumptions were combined with the assumption of constant temperature drop across the condensate film for all the tubes, and the average coefficient for a vertical row of N tubes was predicted to be:

$$\bar{h}_N = 0.725\left[\frac{k^3\rho(\rho-\rho_v)h_{fg}\cdot g}{\mu N\ D\ (T_{sat}-T_w)}\right]^{1/4} \tag{2.2}$$



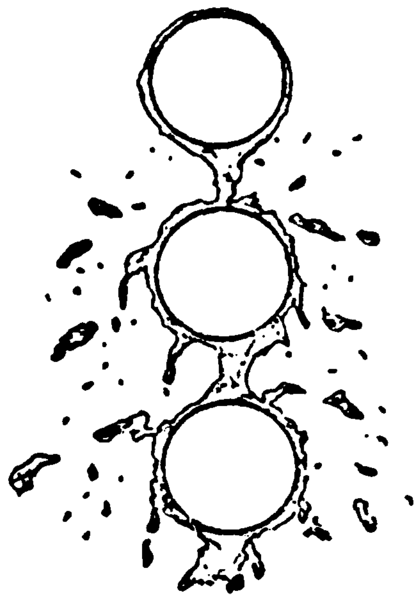


Figure 1a.

Idealized Condensation
on Banks of Tubes

Figure 1b.

More Realistic Picture of
Condensation on Banks of Tubes

Upon, dividing equation (2.2) by equation (2.1), the Nusselt theory can be expressed as:

$$\frac{\bar{h}_N}{h_{Nu}} = N^{-1/4} \tag{2.3}$$

Equation (2.3) can be also expressed in terms of the local coefficient for the N-th tube:

$$\frac{h_N}{h_{Nu}} = N^{3/4} - (N-1)^{3/4} \tag{2.4}$$

In reality, condensate does not drop off in a continuous laminar sheet, but drops off instead by discrete droplets of liquid, as shown in Figure 1b, depending upon the surface tension of the condensate. These droplets create ripples in the condensate film, and thereby decrease the performance degradation due to inundation. Based on his research, Kern [Ref. 20] proposed a less conservative relationship:

$$\frac{\bar{h}_N}{h_1} = N^{-1/6} \tag{2.5}$$

or, in terms of the local coefficient for the N-th tube,

$$\frac{h_N}{h_1} = N^{5/6} - (N-1)^{5/6} \tag{2.6}$$

Chen [Ref. 21] considered the following conditions:

1. the momentum gain of the falling condensate between tubes, and

2. the condensation of vapor on the condensate between tubes,

and concluded that:

$$\frac{\bar{h}_N}{h_{Nu}} = N^{-1/4} \left[(1 + 0.2\zeta(N-1)) \left(\frac{1-0.68\zeta + 0.02\zeta\xi}{1+0.95\xi - 0.15\zeta\xi} \right) \right]^{1/4} \tag{2.7}$$

where;

$$\xi = \frac{k\Delta T}{\mu h_{fg}} \quad , \quad \zeta = \frac{C_p \Delta T}{h_{fg}}$$

and

$$\xi = \frac{\zeta}{P_r}$$

The above approximate expression, *due to Chen, is valid for* most ordinary applications.

Experimental work doen by Eissenberg [Ref. 11], in order to investigate the effects of steam velocity, condensate inundation, and noncondensable gases on the heat-transfer coefficient, revealed that condensate does not always drain onto tubes aligned vertically, but can be diverted sideways, caused by local, vapor-flow conditions. The condensate thus follows a staggered path as shown in Figure 2.

Eissenberg, making the assumption that the flow is dominated by gravity, stated in this side-drainage model, that condensate strikes the lower tubes on their sides

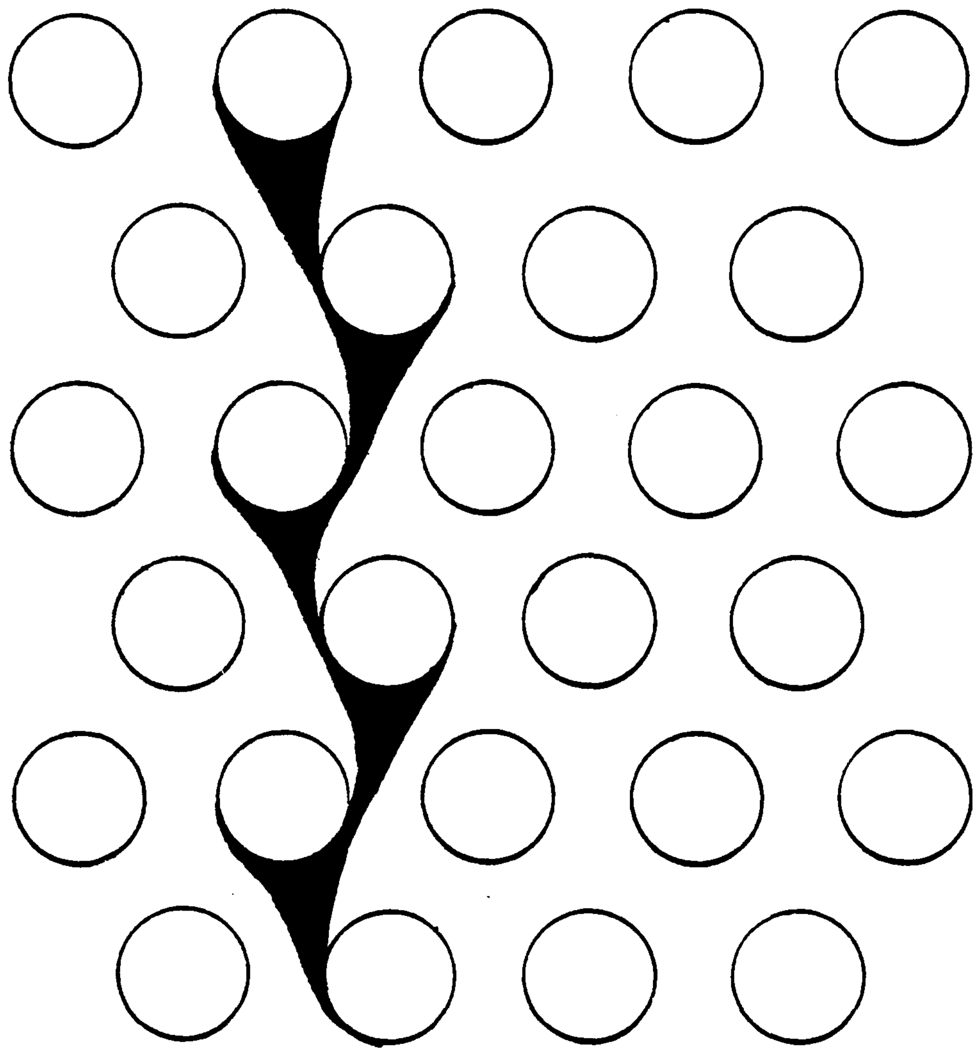


Figure 2. Droplet Path Through a Tube Bundle with Side Drainage

rather than their tops. Therefore, the inundation effects influence the condensate flow only on the lower half of the tubes, which transfer less heat than the upper half. Based on the above-mentioned model, Eissenberg obtained the following formula:

$$\frac{\bar{h}_N}{h_{Nu}} = 0.60 + 0.42\ N^{-1/4} \tag{2.8}$$

Much experimental research has been conducted for studying the effect of condensate inundation. In general, the obtained data are highly scattered as shown in Figure 3. Berman [Ref. 22] made a comprehensive comparison of filmwise condensation data on bundles of horizontal tubes, and he concluded that the wide variation in expermental data for tube bundle inundation is caused by the following variables:

1. bundle geometry (in-line or staggered),
2. tube spacing,
3. type of condensing fluid,
4. operating pressure,
5. heat flux, and
6. local vapor velocity.

In addition, noncondensable gases, and insufficient steam for lower tubes can cause such scattering of data.

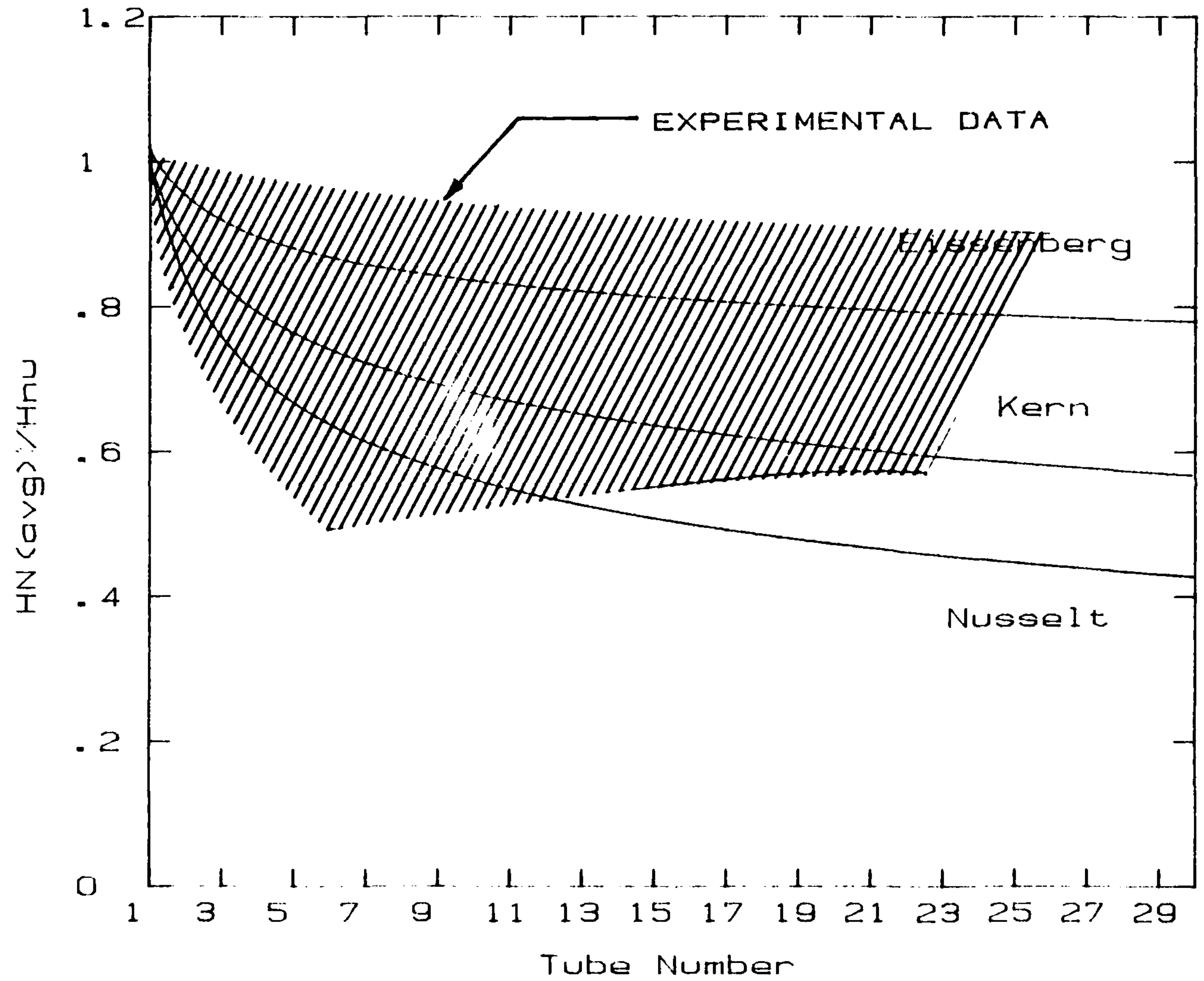


Figure 3. Schematic Comparison of Various Theories with Experimental Data for Condensation Inundation Studies

Nobbs [Ref. 18] used one active tube in a dummy tube bundle. He simulated additional condensate by using three porous tubes. Based on his results, he concluded the following:

1. Vapor velocity increases the condensate heat-transfer coefficient on both inundated and uninundated tubes in a given tube bundle.

2. The effect of inundation is to reduce the heat-transfer coefficient.

3. The condensate drainage path is often not vertically downwards but in a diagonal direction. This can result in tubes receiving different amounts of inundation.

Marto and Nunn [Ref. 23] made a comprehensive survey of the effects of noncondensable gases. They stated that these can be classified into one of two categories:

1. the introduction of an additional local thermal resistance due to the propagation of noncondensable gases towards a condensing tube surface under the influence of a gradient in partial pressure, and

2. the cumulative effect of gas blanketing where uneven rates of condensation in a condenser bundle eventually lead to regions where tubes are inoperative in a condensing role.

Experiments done recently by Cunningham [Ref. 15] revealed that noncondensable gases have a less effect on the performance of finned tubes compared to smooth tubes. Based on the above discussion, it is clear that the performance degradation due to noncondensables must be taken into consideration in realistic designs of tube bundles.

Another very important factor in the performance of a condenser is the effect of vapor velocity. As noted earlier, the vapor shear plays a beneficial role. Berman and Tumanov [Ref. 24] conducted experiments on a single horizontal tube placed in a bank of uncooled neighboring tubes. For vapor in vertical downflow, they found a relation between the vapor Reynolds number, and the heat flux as:

$$\frac{h}{h_N} = 1 + 9.5 \times 10^{-3} \, (Rev)^{11.8/\sqrt{Nu}}$$

with the restriction, that $(Rev)^{11.8/\sqrt{Nu}} < 50$

Eissenberg [Ref. 25] has stated that in designing experimental bundles, the combined effects of inundation and vapor shear are very important. In fact, the use of narrow condenser bundles to experimentally study vapor shear and inundation effects is preferred to simulate large condensers. However, small narrow tube bundles can create errors, due to the following causes:

1. Wall flow: condensate drainage may reach side walls;
2. Dummy tubes: condensate inundation may disperse;
3. Vapor lanes: vapor may bypass along the side walls; and
4. Noncondensable gases: they can affect condensation even at low concentration in the bulk stream, particularly if steam is recycled.

III. EXPERIMENTAL FACILITY

A. TEST FACILITY

The test facility shown in Figures 4 and 5, was designed and built by Morrison [Ref. 13] and modified by Noftz [Ref. 12] to simulate an active tube column having up to 30 tubes in increments of five tubes deep (i.e., five, ten, fifteen, etc.). Some elements of the original test facility were modified to allow for more efficient operation of the facility.

A detailed description of the components used in the test facility is given in Reference 12. Only a short description of these components will be found in this report. Particular attention. however, will be focused on the experimental tubes. Calibration procedures for components requiring calibration are outlined by Reilly [Ref. 7].

B. STEAM SYSTEM

The steam system shown in Figure 6 was modified from Noftz's initial design. From the house supply line, steam flows via a 19 mm O.D. stainless steel line through a steam supply valve (MS-3) to a cast iron steam separator. The steam continues through the system past two Nupro bellows valves, which were used in conjunction with the supply valve

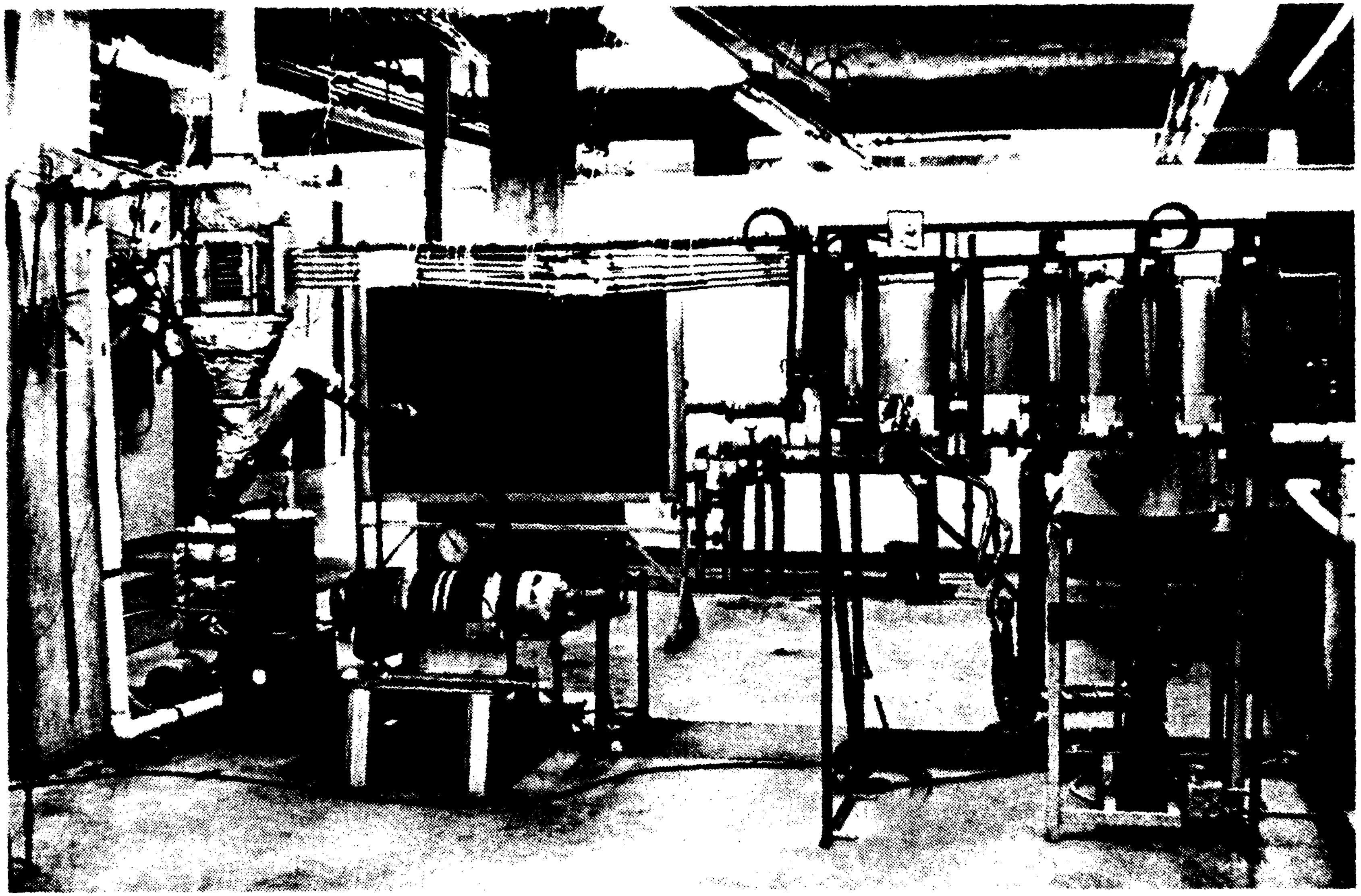


Figure 4. Front View of Test Facility

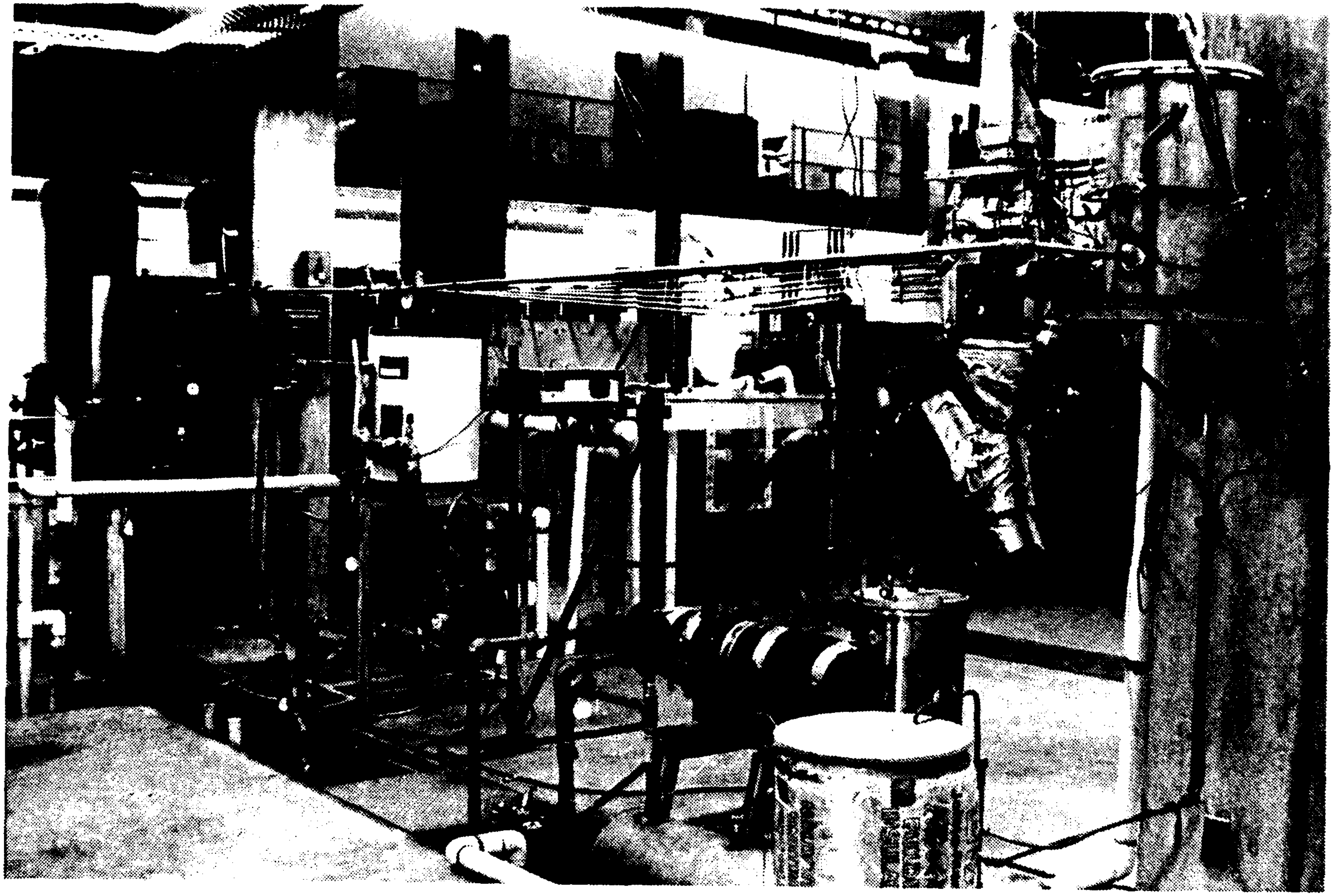


Figure 5. Rear View of Test Facility

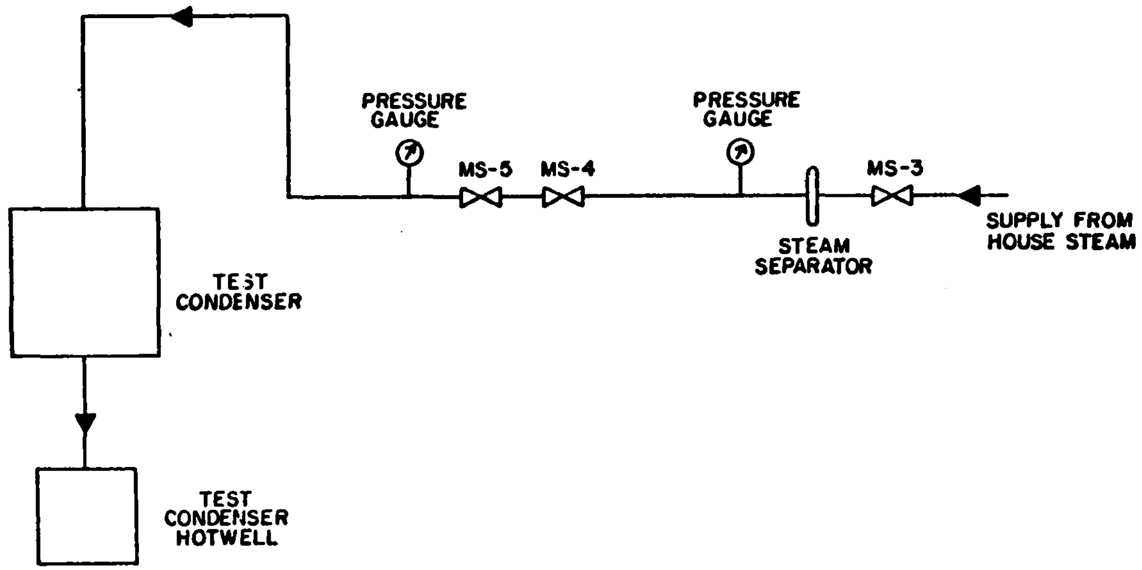


Figure 6. Schematic of Steam System

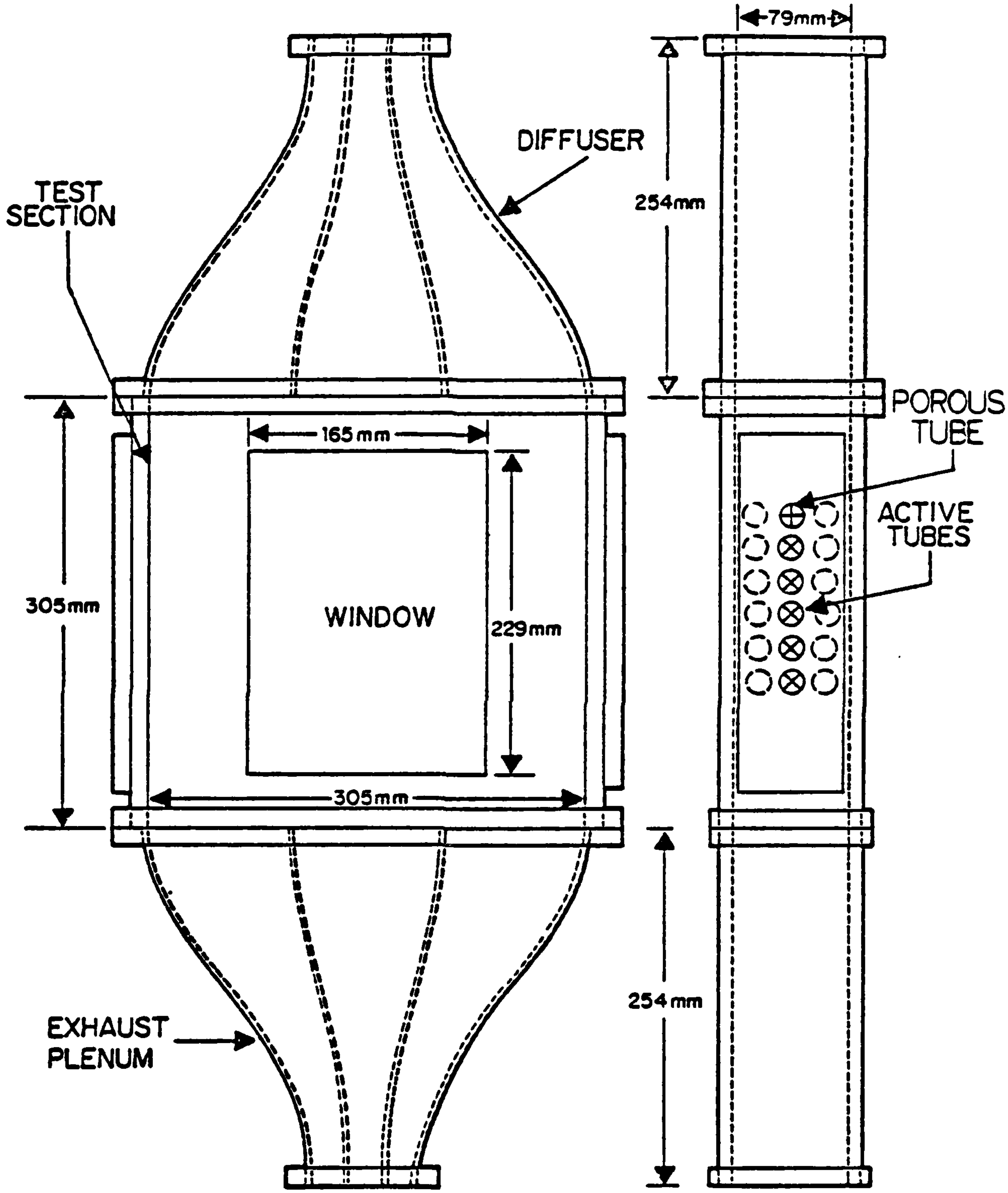
to regulate the steam supply pressure. From these valves, the steam flows into the test condenser diffuser.

The steam supply pressure was monitored by a pressure gage just downstream of the steam supply valve; also a compound gage just after the Nupro valves was used to monitor the pressure in this line.

The operator had no control over the state point, quality or noncondensable gas content since house steam was used. However, especially at nights and during weekends, the state point of the steam at the inlet to the test condenser was found to be nearly constant.

C. TEST CONDENSER

The test condenser shown in Figure 7 was unchanged from Noftz's initial design. Steam enters via the top, passes through the transition piece, the vortex annihilator, and finally through the diffuser. The dimensions of the test condenser were 305 mm X 305 mm X 79 mm and it was made of stainless steel. These dimensions allowed for a maximum of twenty-seven 16 mm O.D. tubes arranged in an in-line configuration of three columns of nine tubes each. For this experiment, the in-line configuration was used with a middle column of five active tubes flanked on either side by a column of five dummy tubes. Just on top of the upper active tube, a perforated, distilled water supply tube was positioned, and flanked on both sides by dummy tubes. In order



NOTE: ALL COMPCNENTS DRAWN TO SCALE

Figure 7. Sketch of Test Condenser

to conduct the experiments, a square, in-line arrangement of tubes, with a pitch-to-diameter ratio of 1.5 was used.

A vertical slot (Figure 7) along the centerline of each condenser end plate was used for active and perforated tube installation. The tubes were positioned using nylon tube sheets that were attached to the exterior of the condenser end plates.

To minimize heat losses, and also to prevent leaks from the tube sheets, each test condenser side was provided with a nylon tube sheet one-inch thick with six holes (S/D = 1.5), having about 0.5 mm tube clearance, which allowed the tubes to be easily slid into the test condenser through the tube sheets. The exterior side of each tube sheet had grooves to support O-rings. The aluminum tube sheets had six holes, having a tube clearance of about 0.5 mm and were used as sealing plates. The diffuser, exhaust plenum, transition piece, vortex annihilator, and the exhaust piping which are shown in Figures 8 and 9, were insulated with rubber insulation.

A viewing window allowed viewing of the condensation process. A double-walled glass window was used, and heated air was fed through the clearance between the two glasses to eliminate fogging on the inner glass.

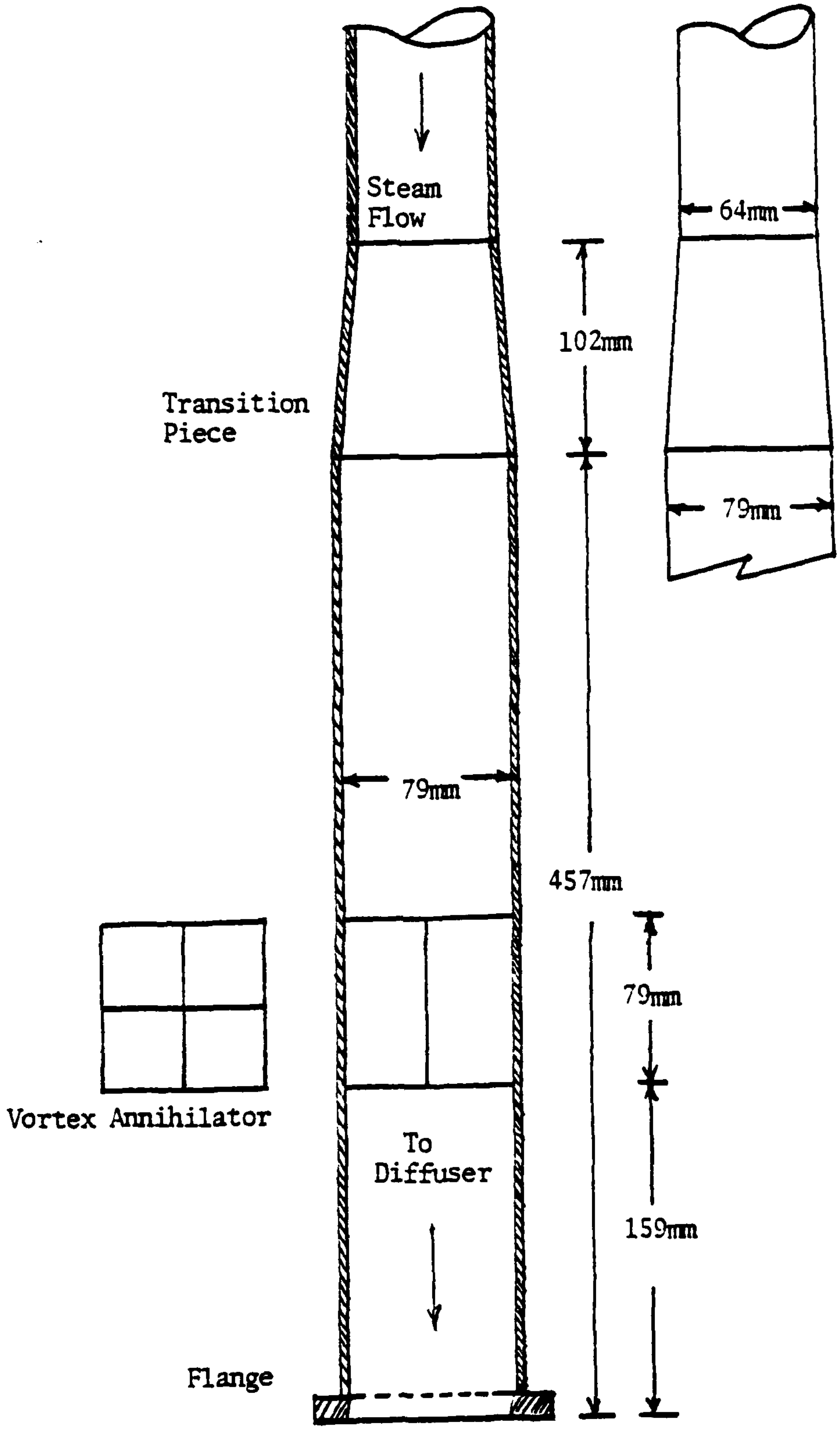


Figure 8. Details of Transition Piece and Vortex Annihilator

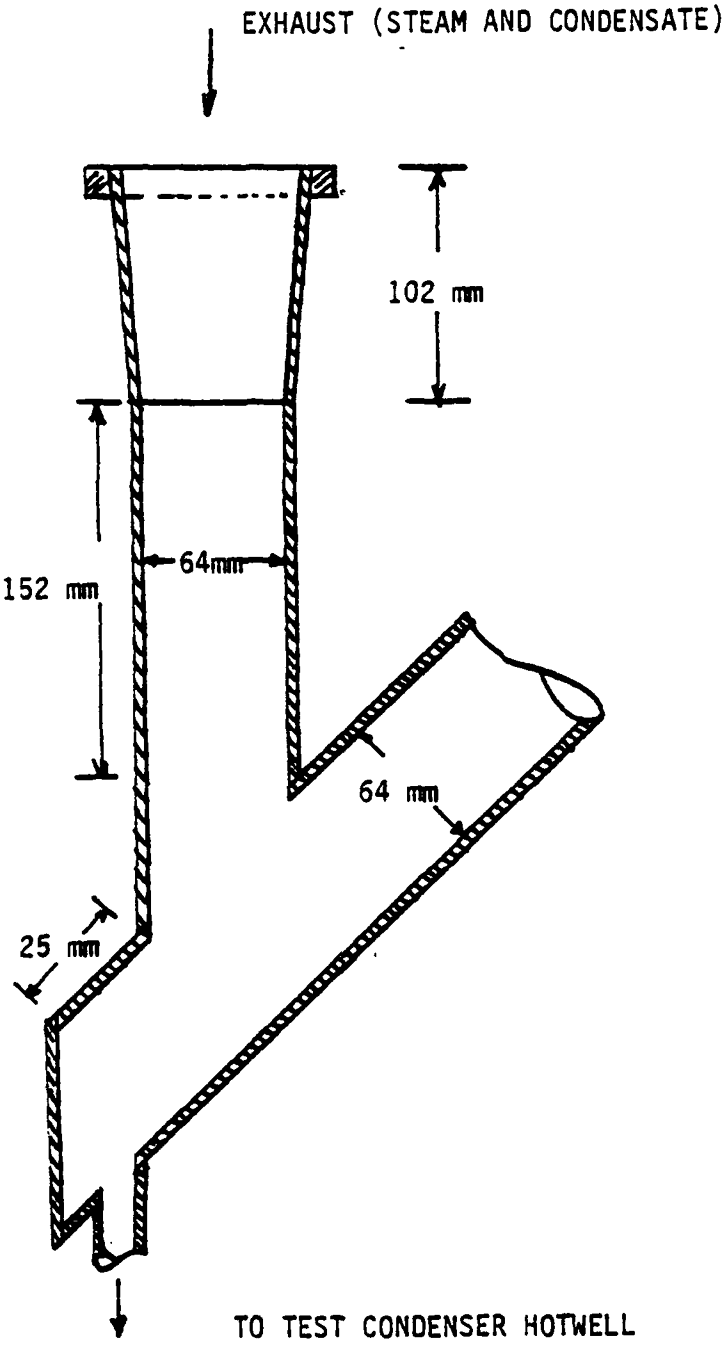


Figure 9. Details of Exhaust and Condensate Piping from Exhaust Plenum

D. TEST CONDENSER TUBES

Four kinds of active tubes were tested in this experiment. The tubes were manufactured by the Wolverine Division of Universal Oil Products, and all were made of titanium.

The first kind of tubes, shown in Figure 10 were smooth titanium tubes of 16 mm O.D. with a 1.65 mm wall thickness. The second kind of tubes, shown in Figure 11, were single-start, helically-corrugated tubes, designated as Low Pressure Drop (LPD), with 16 mm O.D. and a 1.65 mm wall thickness. The third kind of tubes, shown in Figure 12, were also single-start, helically-corrigated LPD tubes. wrapped with titanium wire of 1.58 mm O.D. The fourth kind of tubes, shown in Figure 13, were smooth tubes, wrapped with 1.58 mm O.D. titanium wire on the same pitch as the roped Wolverine tubes. In order to wrap the titanium wire, the tubes were attached into a lathe, and the wire was welded at the start of the helical groove. Then, under a constant tension of 4 kg, the wire was wrapped manually by turning the chuck of the lathe. Finally, the free end of the titanium wire was welded at the end of the helical groove.

The 660-mm-long active tubes were connected to separate cooling water supply and discharge lines. The dummy tubes flanking the active tubes were made of 16 mm O.D. stainless steel. These tubes served to direct the steam flow so as to

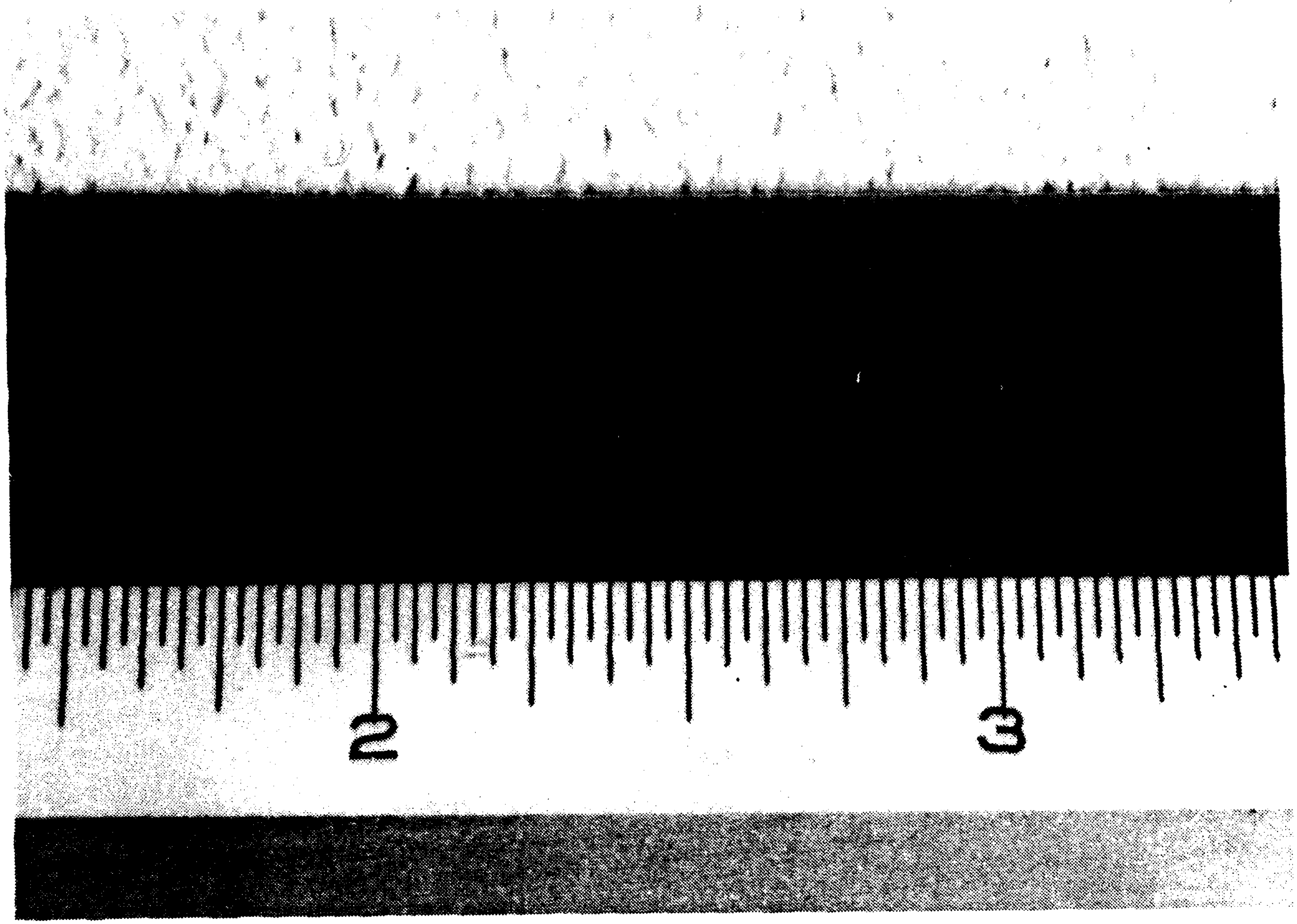


Figure 10. Photograph of Smooth Titanium Tube

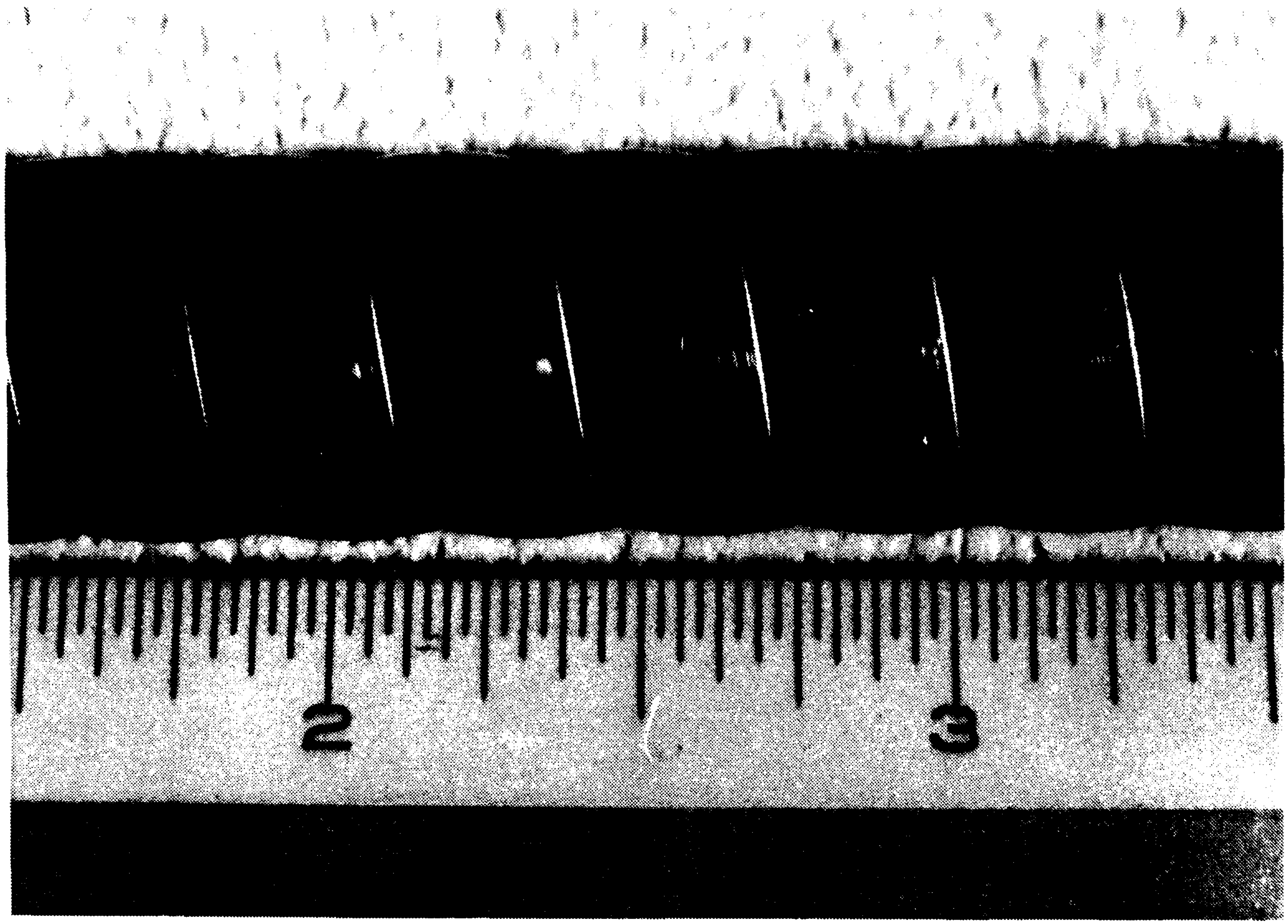


Figure 11. Photograph of Roped Titanium Tube

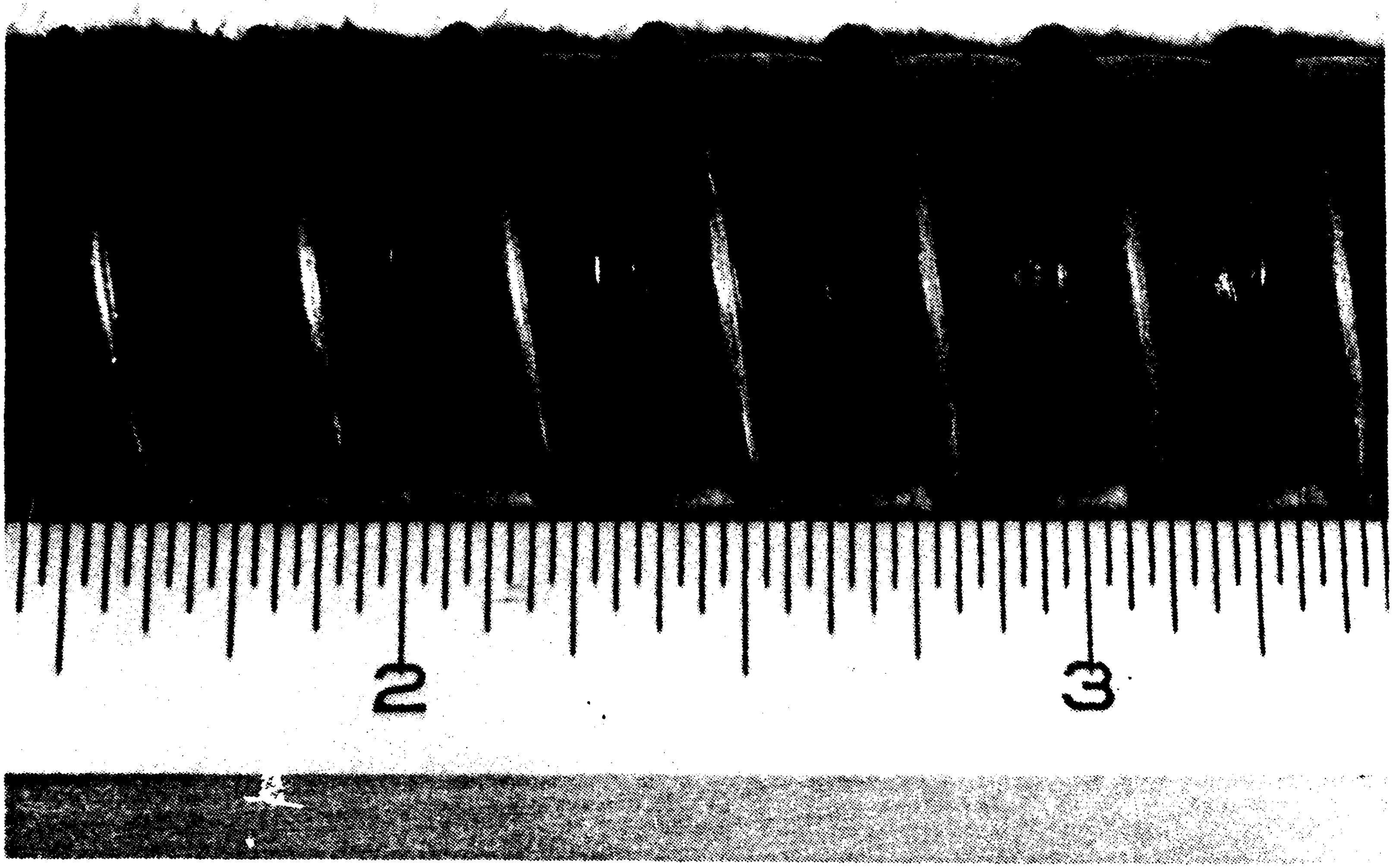


Figure 12. Photograph of Wire Wrapped Roped Titanium Tube

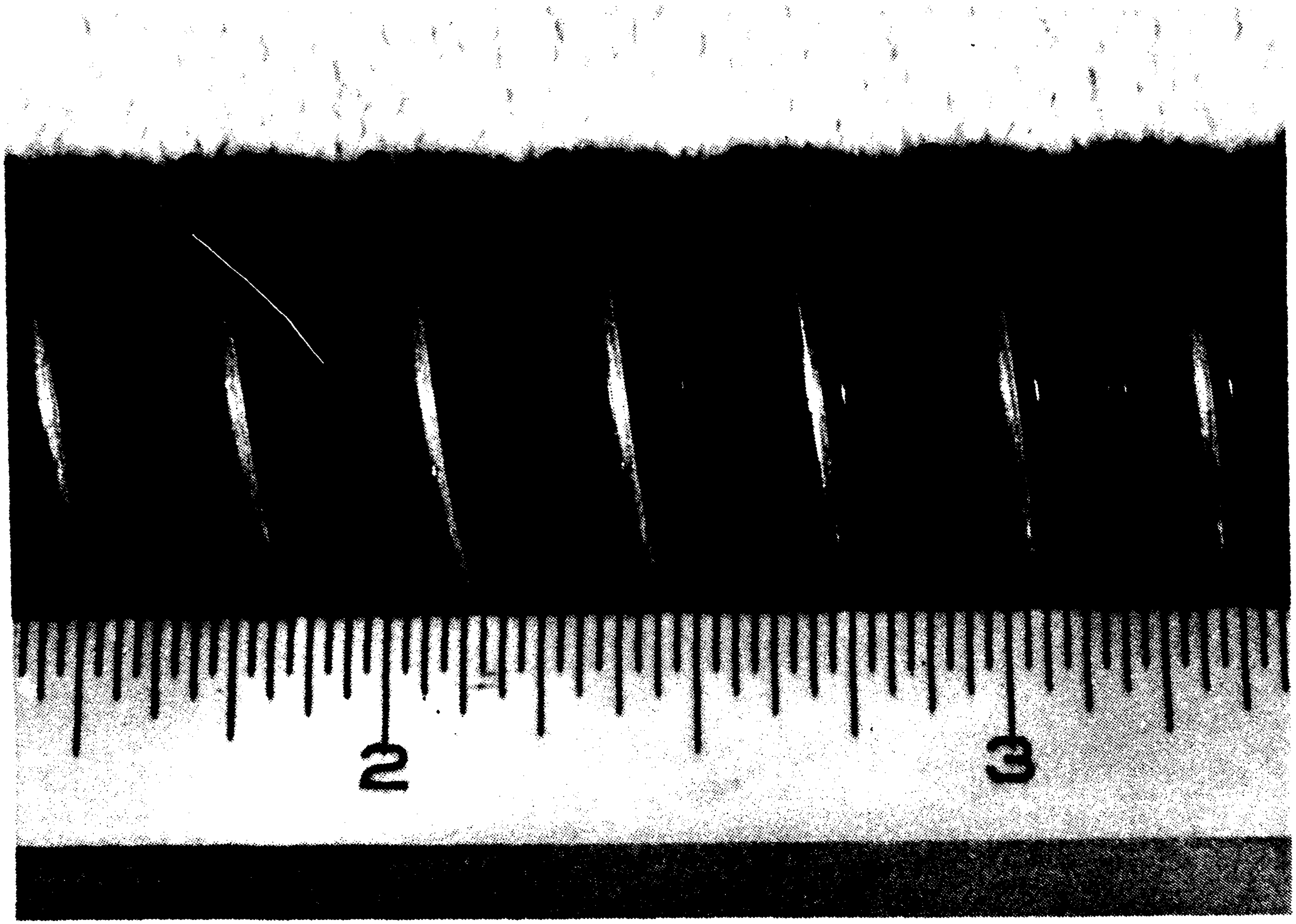


Figure 13. Photograph of Wire Wrapped Smooth Titanium Tube

simulate actual conditions in a condenser. Cooling water was not supplied to these tubes and they did not penetrate the test condenser end plates. Special characteristics of the Wolverine tubes are listed in Table I.

E. PERFORATED TUBE

The perforated tube water supply system is shown in Figure 14. This system consisted of a perforated tube (located above the uppermost active tube), a water heater which served as a supply tank, a rotameter in order to control the amoung of water supplied to the perforated tube, a pump driven by a 1/2-HP electric motor, a condensate pump and associated piping system and valves. The active length of the copper-nickel, perforated tube was 305 mm, which was identical to the length of the test condenser. Supply water entered one end of this tube; the other end was sealed off.

The supply tank was unchanged from Noftz's initial design, except for the addition of a thermocouple in order to have a direct indication of the exact temperature of the water supplied to the perforated tube. The water, heated to the temperature of the condensate leaving the bottom tube, was fed to the perforated tube by a 1/2-HP, electric-motor-driven pump, via a rotameter and a recirculation valve. The flow rate to the perforated tube was controlled by using the rotometer and the valve provided in the heater-water recirculation line. Also, another modification was made to

TABLE I

DESIGN CHARACTERISTICS OF WOLVERINE CORODENSE TUBES (TYPE LPD)

Outside diameter (in)	: 5/8
Wall thickness (in)	: 0.035
Outside diameter (ft)	: 0.0520
Heat Transfer Surface and Ratio	
a. Outside A_o^* (ft^2/ft)	: 0.163
b. Outside to inside A_o^*/A_i^*	: 1.141
Inside diameter D_i (ft)	: 0.0435
Cross section for flow A_{cs} (ft^2)	: 0.00163
Number of groove starts	: one ± 0
Pitch (in)	: 0.300
Groove radius (in)	: 1/32 nominal
Depth-transition length (next to plain section) (in)	: 1-3/4 max

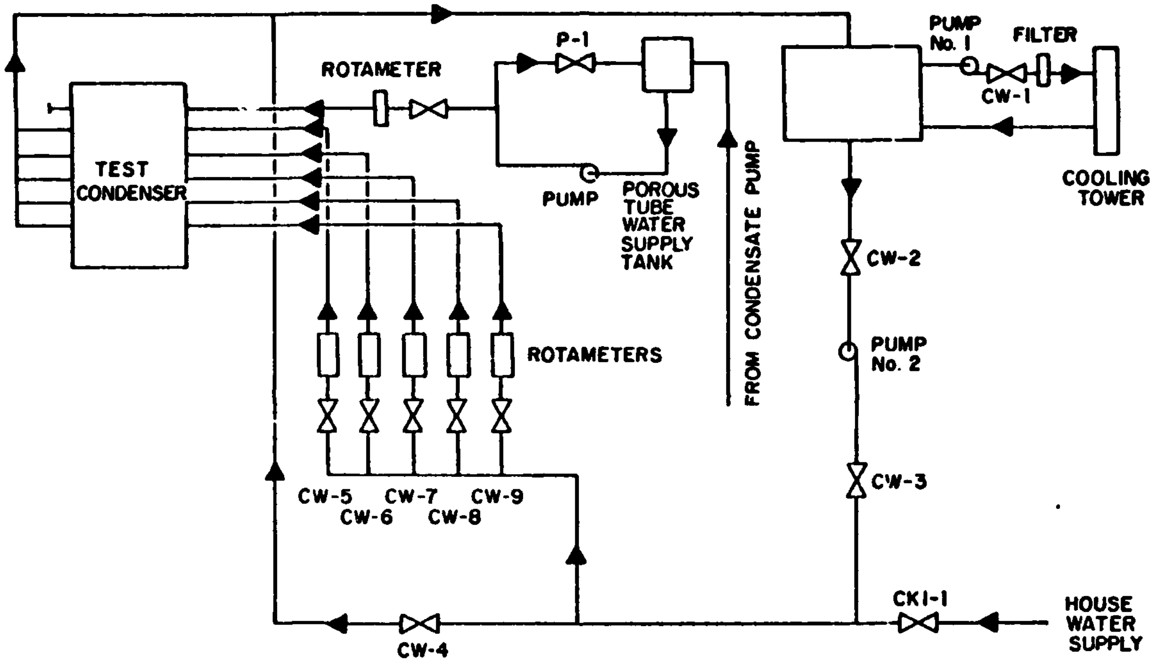


Figure 14. Schematic of Perforated Tube Water Supply System

feed the condensate collected in the hotwell, via the condensate pump, to the supply tank (heater) in order to facilitate the inundation of up to 30 tubes under atmospheric conditions.

The main feature of the perforated tube was to inundate with condensate from above, in order to simulate a tube column of more than five active tubes. When the wrapped Wolverine tubes and the smooth wrapped tubes were tested, the perforated tube was also wrapped with titanium wire, by using the above-mentioned technique.

F. CONDENSATE SYSTEM

The condensate system shown in Figure 15 was modified from Noftz's initial design, and consisted of the test condenser and hotwell, the condensate pump, piping and valves.

The test condenser hotwell collected the condensate produced by the test tubes. With valve C-1 closed, the condensate mass flow rate from the test condenser could be measured using a stop watch. Opening the valve, the condensate was fed, via the condensate pump, to the supply tank for the perforated tube, or dumped into the building's drainage system. The condensate line connecting the test condenser and the test condenser hotwell were insulated using rubber insulation.

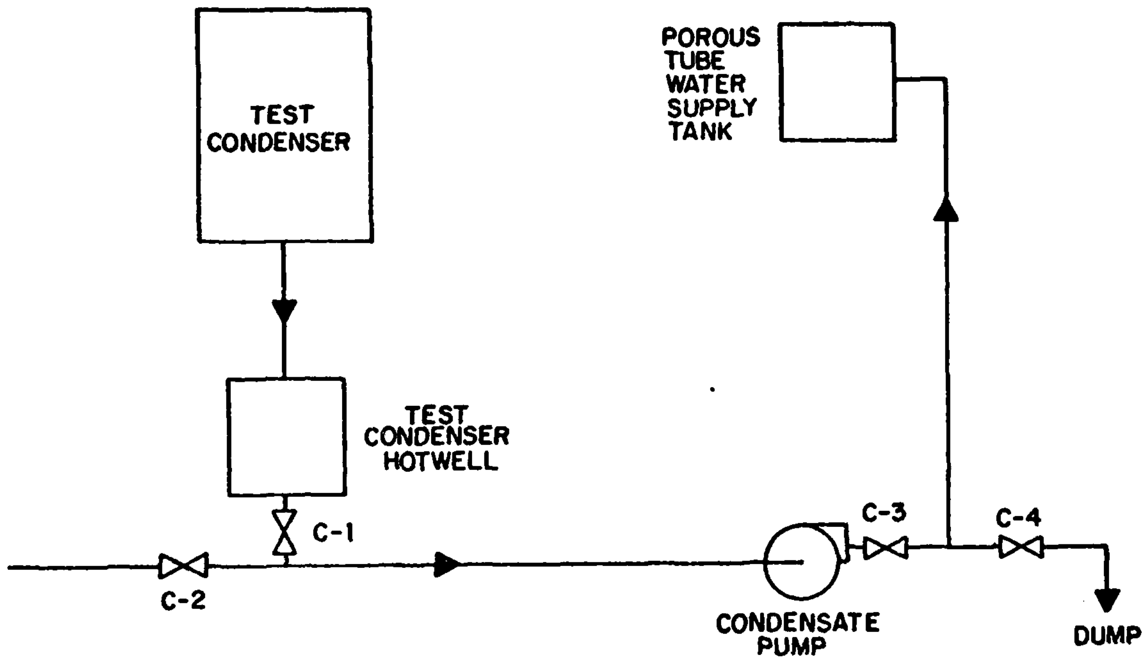


Figure 15. Schematic of Condensate System

G. COOLING-WATER SYSTEM

The cooling-water system was a partially-closed system as shown in Figure 14, and it was unchanged from Noftz's initial design. House water was used for the test facility. Cooling water was stored in a 1.2 meter, cubical, plexiglass supply tank, and was pumped by a 5-HP, electrically-driven pump via 51-mm-O.D. plastic piping to a manifold. Five rotameters were attached to the manifold to measure the flow rate through each active tube. Five regulating valves were used to obtain any water velocity between Ø and 5 m/s within the tubes. The cooling water passing through the rotameters was fed via 16 mm O.D. stainless steel tubing, and tygon tubing to the active tubes. The total tube lengths were long enough (over 2 meters) to ensure hydraulically fully-developed flow, and no swirling into the tubes. After flowing through the tubes, the cooling water was passed through mixing chambers, where the temperature profile was destroyed, to facilitate the steady measurement of the outlet cooling water temperatures. After the mixing chambers, the cooling water was collected in the supply tank. A 7.5-HP, electrically-driven pump was used to pump water from the supply tank, through a filter, to a cooling tower in order to minimize temperature rise at the water inlet.

The cooling tower was located outside the building and was composed of four truck radiators across which air was blown by means of a fan. The entire system, consisting of the heat exchanger and the fan was enclosed in a wooden structure with louvered openings to provide enough ventilation.

The tygon tubing was secured to the inlet and outlet sections of the active tubes by means of hose clamps. The outlet sections, including the mixing chambers, were insulated with rubber insulation.

H. INSTRUMENTATION

1. Flow Rates

a. Foulton rotameters were used to measure the flow rate of cooling water for each active tube. Starting from the top active tube to the bottom one, the rotameters were calibrated giving 100% maximum flow rates of 66.9±1, 72.6±1, 71.5±1, 72.8±1, and 73.3±1 kg/min.

b. The perforated tube water rotameter was calibrated giving a 100 percent volumetric flow rate of 969±30 ml/min.

c. All rotameters were calibrated using the procedures noted in Appendix A of Reference 7.

2. Temperature

Stainless-steel sheathed, copper-constantan thermocouples were used as the primary temperature monitoring devices. Three thermocouples were used to measure the inlet cooling water temperature; and twelve thermocouples were utilized to measure the outlet cooling water temperature.

Additionally, two thermocouples were used to measure the steam saturation temperature; one thermocouple was used to monitor the condensate temperature, and one was used to measure the vapor temperature. Table II lists the locations monitored.

A Gulton Industries, West 20, 0-500° F temperature controller was used to regulate the temperature of the perforated tube supply water. The controller had a manufacturer-stated accuracy of ±0.5% of the span (about ±1.25° F). It was impossible to obtain the precise temperature for the perforated tube water, because of the low response of the heating system.

During inundation runs, the condensate temperature rose with the increasing number of tubes. For example, the condensate temperature after the 5th tube was 94.6° C and it increased to 97.65° C after the 30th tube.

3. <u>Pressure</u>

Several different types of pressure measurement devices were used in this facility. They were: a Bourdon tube pressure gage which was used to measure the steam supply pressure, located downstream of the steam supply valve; an absolute pressure transducer which was used to measure the test condenser pressure; and, a compound pressure gage was also used to measure the pressure drop downstream of the Nupro valves. The pressure transducer was calibrated against a mercury manometer.

4. <u>Data Collection and Display</u>

A Hewlett-Packard HP-3Ø54A Automatic Data Acquisition System, with a HP-9826 computer and a HP-2671G printer were used to record and display the thermocouple and pressure transducer readings. The temperatures were recorded in degrees Celsius, while the pressure was recorded in volts which was then converted, using the calibration curve, into mm Hg absolute. The pressure transducer was assigned to channel Ø19 of the HP Automatic Data Acquisition System, while the thermocouples were assigned channels as indicated in Table II.

TABLE II

CHANNEL NUMBERS FOR COPPER-CONSTANTAN THERMOCOUPLES

Location	Channel
T_{ci} #1	000
T_{ci} #3	001
T_{ci} #5	002
T_{co} #1	003
T_{co} #1	004
T_{co} #1	005
T_{co} #2	006
T_{co} #2	007
T_{co} #3	008
T_{co} #3	009
T_{co} #4	010
T_{co} #4	011
T_{co} #4	012
T_{co} #5	013
T_{co} #5	014
T_{sat}	015
T_{sat}	016
T_{cond}	017
T_{vap}	018

IV. PROCEDURES

A. INSTALLATION AND OPERATING PROCEDURES

1. Preparation of Condenser Tubes

Prior to installation, the titanium tubes were cleaned using a chemical cleaning method [Ref. 26]. The steps in this cleaning process are as follows:

a. Swab the tube surface with acetone to remove grease.

b. Using a test tube brush, brush the inside surface of the tube with a 50% sulfuric acid solution in order to remove any oxides. Also, apply this solution to the outside surface of the tube.

c. Rinse the inside and outside of the tube with tap water.

d. Apply, using a brush, a 50% solution of sodium hydroxide mixed with an equal amount of ethyl alcohol, at the boiling temperature (about 85° C) to the outside surface of the tube.

e. Rinse the tube with tap water.

f. Rinse thoroughly with distilled water.

Prior to any run, the condenser tubes had to be prepared to ensure filmwise condensation. Exterior and interior surfaces were cleaned to ensure proper wetting characteristics. Also, the tubes were cleaned by running steam at atmospheric pressure through the test condenser for about twenty minutes without the cooling water running through the tubes.

It was also found that, when drop-wise condensation occurred, rinsing the tubes using the perforated-tube water was sufficient to restore film-wise condensation.

2. System Operation and Steady-State Conditions

Complete operating instructions are listed in Appendix A. The steady-state condition was reached about three hours after initial system light-off, and about fifteen minutes after changes to the cooling water or perforated tube supply water flow.

When a steady-state condition was reached. the runs were made. The duration of each run was approximately one minute. For each condition, for example, for tubes 11 through 15, five consecutive runs were made and average values were computed.

The following data were taken automatically by the data-acquisition system:

a. the thermocouple readings, and

b. the pressure transducer reading.

Also, the following data were read into the computer, through the keyboard, for each run:

a. the setting of each rotameter, and

b. the test condenser hotwell levels.

B. DATA REDUCTION PROCEDURES

1. Overall Heat-Transfer Coefficient

The heat-transfer rate to the cooling water is given by:

$$Q = \dot{m}C_{pw}(T_{co}-T_{ci}) \tag{4.1}$$

The heat-transfer rate can also be found from the overall heat-transfer coefficient by:

$$Q = U_o \; A_o \cdot \text{LMTD} \tag{4.2}$$

where

$$\text{LMTD} = \frac{(T_{sat}-T_{ci}) - (T_s-T_{co})}{\ln\left(\frac{T_{sat}-T_{ci}}{T_{sat}-T_{co}}\right)} \tag{4.3}$$

After combining equations (4.1), (4.2) and (4.3), it is found that

$$U_o = \frac{C_{pw}}{A_o} \ln\left(\frac{T_{sat}-T_{ci}}{T_{sat}-T_{co}}\right) \tag{4.4}$$

2. Inside Heat-Transfer Coefficient

The heat-transfer coefficient on the inside is calculated from the Sieder-Tate relationship as described in Holman [Ref. 27]:

$$N_u = \frac{h_i D_i}{k} = C_i \, R_e^{0.8} \cdot P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (4.5)$$

In the above equation. C_i is referred to as the Sieder-Tate coefficient. The remainder of the right-hand side of the above equation,

$$\left(R_e^{0.8} \cdot P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}\right)$$

is referred to as the Sieder-Tate parameter, X.

For thermally and hydrodynamically developed flow in tubes, the Sieder-Tate coefficient equals 0.027. However, when short tubes are considered, as in the case of this experiment, fully-developed conditions are not attained. Therefore, the value of C_i must be found experimentally.

The Wilson plot was used to arrive at the value of the Sieder-Tate coefficient. The Wilson plot was developed in 1915 [Ref. 28]. It is merely a plot of $1/U_o$ versus the inverse of the Sieder-Tate parameter (which is proportional to the inverse of cooling water velocity raised to the 0.8 power). The reasoning behind the Wilson plot is shown in the following development.

Consider the equation for the overall heat-transfer coefficient:

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + R_w + \frac{1}{h_o} \tag{4.6}$$

or

$$\frac{1}{U_o} = \frac{A_o}{A_i} \frac{D_i}{C_i k X} + R_w + \frac{1}{h_o} \tag{4.7}$$

where

$$R_w = \frac{D_o \ell n(\frac{D_o}{D_i})}{2k_m}$$

For the Wilson-plot method to be successful, h_o must be kept constant. When steam condenses on the shell side, the above condition can be achieved only if the heat flux (q") is kept constant.

As required for the Wilson plot. when the cooling water velocity increases. h_i increases. U_o increases, and finally q" increases.

Consider

$$q'' = U_o \mathrm{LMTD} \tag{4.8}$$

In order to keep q" constant, one must therefore decrease LMTD by lowering the steam saturation temperature. This requires trial-and-error setting of condenser pressure which is a very difficult task.

To avoid this difficulty and yet arrive at a Sieder-Tate constant not affected by the varying q", a modified Wilson-plot method was used. This method was developed by Wanniarachchi [Ref. 29], and the required steps are listed below:

1. Assume $C_i = 0.027$
2. Calculate

$$LMTD = \frac{T_{co} - T_{ci}}{\ln\left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}}\right)}$$

3. Calculate

$$Q = \dot{m} C_p \, (T_{co} - T_{ci})$$

4. Calculate

$$U_o = \frac{Q}{A_o \cdot LMTD}$$

5A. For the first data point only, do the following:

a. Assume

$$C_f = \left(\frac{\mu}{\mu_u}\right)^{0.14} = 1$$

b. Calculate the Sieder-Tate parameter

$$X = R_e^{\,0.8} \, P_r^{\,1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

c. Calculate

$$h_i = \frac{k}{D_i} C_i \, R_e^{\,0.8} \, P_r^{\,1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

d. Calculate

$$T_w = T_c + \frac{q''}{h_i}$$

e. Calculate

$$C_f = \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

f. Repeat steps a through e until C_f assumed in step a approximately equals C_f calculated in step e.

g. Calculate

$$\frac{1}{h_o} = \frac{1}{U_o} - \frac{1}{h_i}\frac{A_o}{A_i} - R_w \tag{4.9}$$

h. Assign $Q_1 = Q$

NOTE: The second subscript of h on the left-hand side of equation (4.9) refers to the first data point.

5B. For all data points <u>except</u> for first data point. do the following:

a. Calculate

$$\frac{1}{h_i} = \left[\frac{1}{U_o} - R_w - \frac{1}{h_{o,N}}\right]\frac{A_i}{A_o}$$

where

$$\frac{1}{h_{o,N}} = \frac{1}{h_{o,1}} - \left(\frac{Q}{Q_1}\right)^{1/3}$$

b. Calculate

$$T_w = T_c + \frac{q''}{h_i}$$

c. Calculate the Sieder-Tate parameter

$$X = R_e^{0.8} P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

6. Repeat steps 2 through 5 for all data points.

7. Plot $\frac{1}{h_i}$ versus $\frac{1}{X}$

NOTE: It is more reasonable to plot h_i versus X. However, $1/h_i$ versus 1/X was plotted in order to be consistent with the original Wilson-plot method

$$\left(\frac{1}{U_o} \text{ versus } \frac{1}{X} \right).$$

8. Obtain the slope (m) of the least-squares-fit straight line.

9. Calculate the Sieder-Tate constant.

$$C_i = \frac{D_i}{k_w \cdot m}$$

10. Repeat steps 2 through 9 until the assumed and calculated C_i values are approximately equal.

A listing of the computer program used in this method can be found in Appendix D.

3. <u>Outside Heat-Transfer Coefficient</u>

The heat-transfer coefficient on the outside is calculated using the following steps:

1. Calculate the average bulk water temperature.

$$T_b = (T_{co} + T_{ci})/2$$

2. Evaluate the thermophysical properties based on the average bulk water temperature.

3. Calculate the cooling water velocity.

$$V_w = \frac{\dot{m}}{\rho A_i}$$

4. Calculate the water-side Reynolds number.

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

5. Calculate the heat transferred to the cooling water.

$$Q = \dot{m}\ (T_{co} - T_{ci})\ C_{pw}$$

6. Calculate the heat flux.

$$q'' = \frac{Q}{\pi \cdot D_o \cdot L}$$

7. Calculate the Nusselt coefficient using the formula:

$$h_{Nu} = 0.651 \left[\frac{k_f^{\,3} \cdot \rho_f^{\,2}\ h_{fg} \cdot g}{\mu_f \cdot D_o \cdot q''} \right]^{1/3} \qquad (4.10)$$

NOTE: The above formula was employed since no direct measurement of tube wall temperature was made.

To compare the condensate film temperature, as required for equation (4.9), an iterative scheme was used as outlined below:

a. Assume $T_f = T_{sat}$.

b. Evaluate the relevant thermophysical properties. which are included in equation (4.10).

c. Calculate the Nusselt coefficient using equation (4.10).

d. Evaluate $T_{f,c}$ using the formula:

$$T_{f,c} = T_{sat} - \frac{q''}{h_{Nu}} + 0.5$$

e. Repeat steps a through e until T_f assumed in step a approximately equals $T_{f,c}$ calculated in step e.

8. Calculate the inside heat-transfer coefficient using the formula:

$$h_i = \frac{k_w}{D_i} C_i \, R_e^{\,0.8} \, P_r^{\,1/3} \cdot C_f \qquad (4.11)$$

NOTE: In order to determine the C_f, the average inner wall temperature must be known. As noted earlier, this temperature is not known directly and it must be found iteratively, as described below:

a. Assume a correction factor C_f (say 1).

b. Calculate the h_i using equation (4.11).

c. Calculate the tube wall temperature using the formula:

$$T_w = T_b + \frac{q''}{h_i} \left(\frac{D_i}{D_o}\right)$$

d. Calculate a new correction factor, at the evaluated tube wall temperature

$$C_f = \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

e. Repeat steps a through d until C_f assumed in step a approximately equals C_f calculated in step d.

9. Calculate the log-mean-temperature difference, (LMTD).

$$LMTD = \frac{T_{co} - T_{ci}}{\ln\left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{c.}}\right)} \tag{4.12}$$

10. Calculate the overall heat-transfer coefficient using the formula:

$$U_o = \frac{q''}{LMTD} \tag{4.13}$$

11. Calculate the outside heat-transfer coefficient using the equation:

$$h = \frac{1}{\frac{1}{U_o} - \frac{D_o}{D_i h_i} - R_w} \tag{4.14}$$

NOTE: The validity of equation (4.13) is based on the assumption of negligible water-side fouling resistance and the resistance due to noncondensables.

12. Calculate the normalized, local, outside heat-transfer coefficient, h_N/h_1.

13. Calculate the normalized average, local outside heat-transfer coefficient $(\bar{h}_N/h_1)$.

C. DATA-REDUCTION PROGRAM

A computer program was utilized to analyze the raw data. The program was in BASIC language and was run on an HP-9826 computer system. A peripheral plotter was used to plot the results. The program is presented in Appendix D.

V. RESULTS AND DISCUSSION

A. SIEDER-TATE COEFFICIENTS FOR SMOOTH AND ROPED TUBES

The modified Wilson Plots for smooth and roped tubes are shown in Figures 16 and 17. Of particular interest is the very small intercept, which represents the estimated fouling factor. For example, the estimated fouling factor for data run STSD-11 (for smooth tubes) is 1.37×10^{-6} $m^2 \cdot K/W$ based on the inside area. This value is only three percent of the typical value (4.4×10^{-5} $m^2 \cdot K/W$) used for design purposes. This very small value supports the initial assumption of negligible fouling factor for the success of the modified Wilson-Plot method.

Table III is a summary of the Sieder-Tate coefficients calculated for smooth and roped tubes. The average Sieder-Tate coefficient was calculated to be 0.029 ± 0.001 for smooth tubes and $0.061 + 0.002$ for roped tubes. Thus, the roped tubes have a Sieder-Tate coefficient 2.1 times greater than that for the smooth tubes. This increase is mainly due to the increased surface area, turbulence and swirl effects.

The Sieder-Tate coefficient derived for the smooth tubes is 7.4 percent greater than the value (0.027) published in the original, generalized correlation. This increase can be easily explained by the short condensing tubes used in this

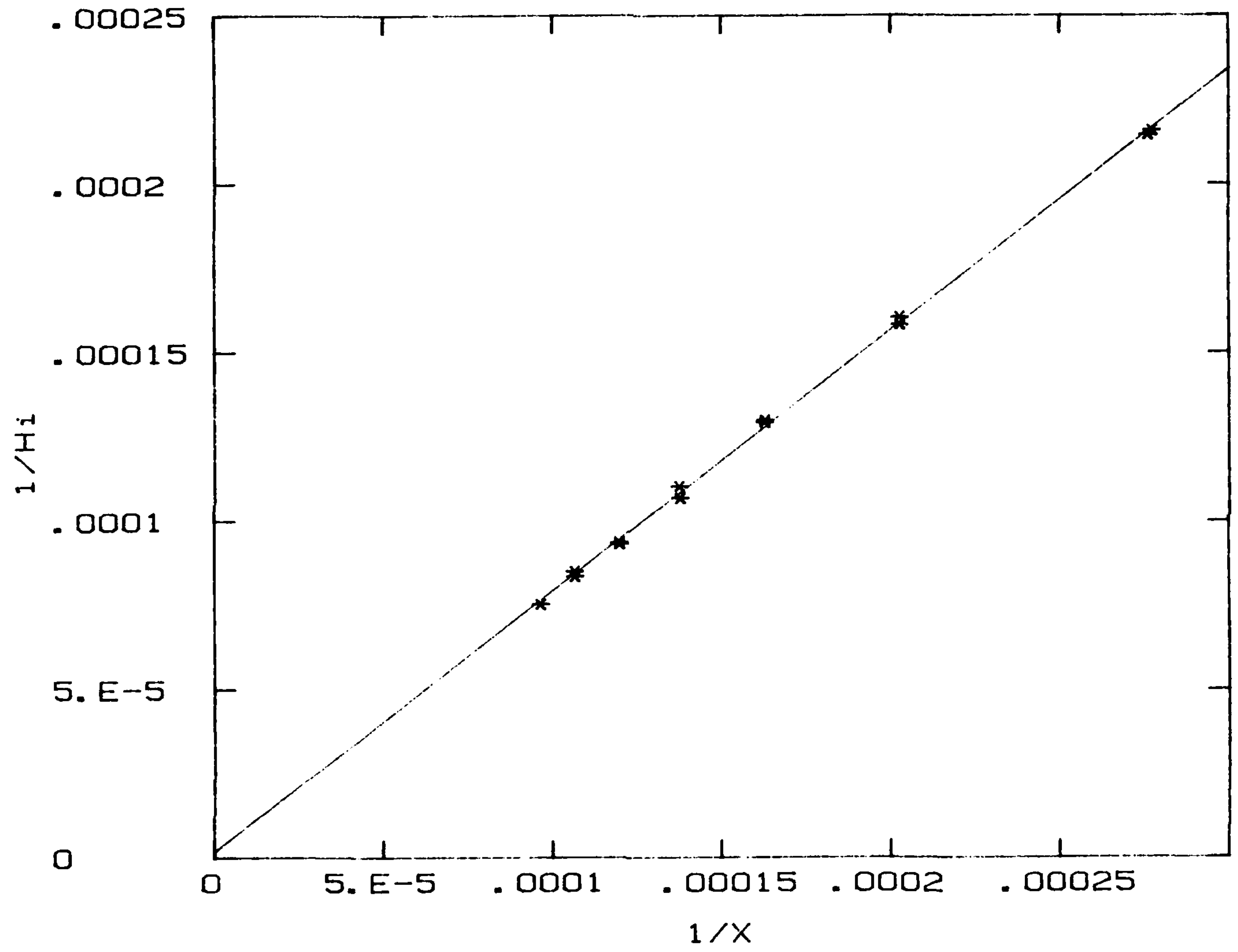


Figure 16. Modified Wilson-Plot for Smooth Tubes (Run STSD-11)

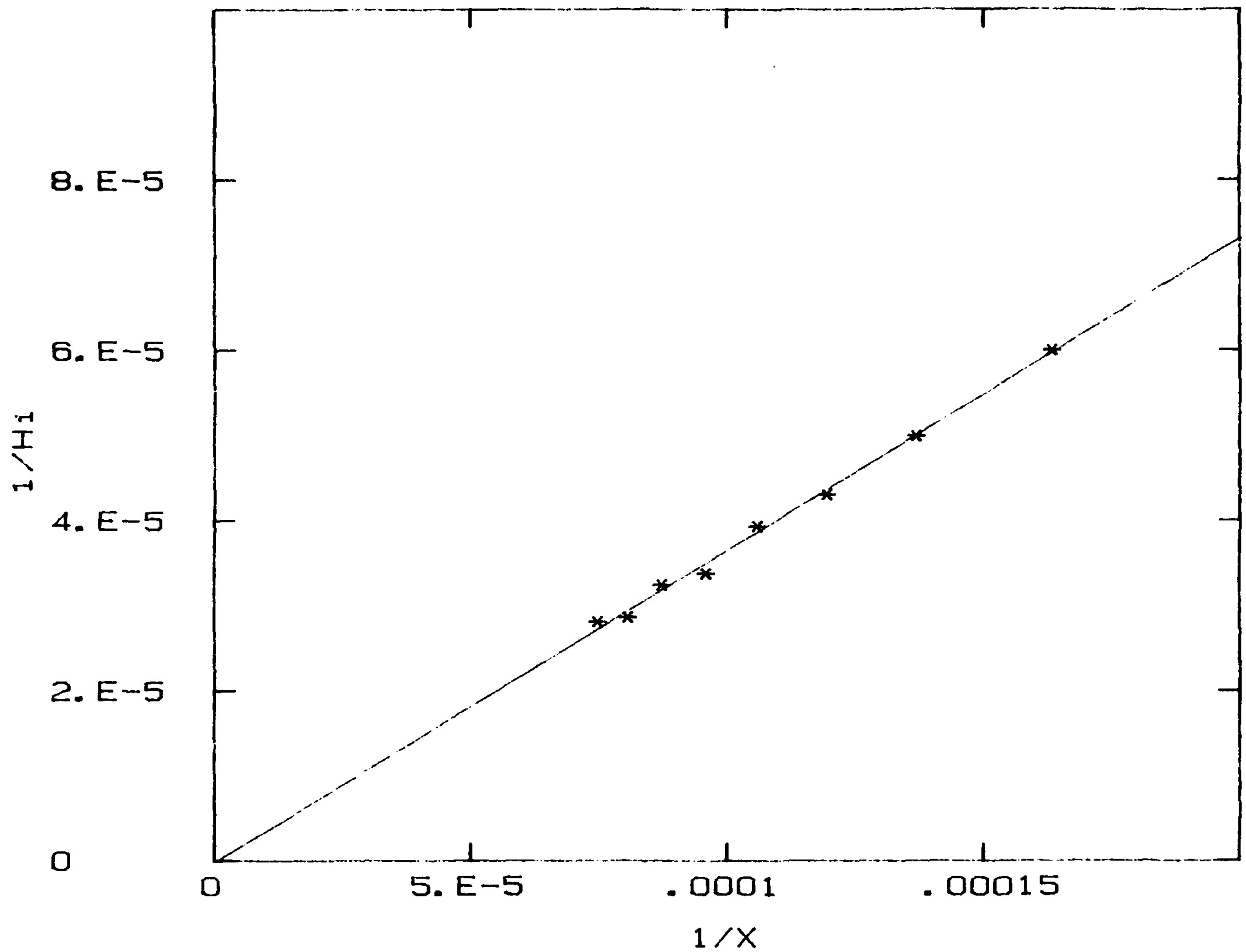


Figure 17. Modified Wilson-Plot for Roped Tubes (Run RTSD-11)

TABLE III

SUMMARY OF SIEDER-TATE COEFFICIENTS
FOR SMOOTH AND ROPED TUBES

File Name	Tube Type	Sieder-Tate Coefficient
STSD-2	Smooth	0.0287
STSD-10	Smooth	0.0291
STSD-11	Smooth	0.0296
RTSD-3	Roped	0.0591
RTSD-4	Roped	0.0589
RTSD-6	Roped	0.0621
RTSD-7	Roped	0.0627

experiment. Even though the flow condition was hydrodynamically fully-developed, it was thermally developing throughout the condensing length, resulting in a greater Sieder-Tate coefficient.

It is worth noting that the Sieder-Tate coefficient derived by using the original. Wilson-Plot method (i.e., plotting $1/U_o$ versus $1/X$), consistently gave values about 10% greater than the values obtained using the modified. Wilson-Plot method.

B. SMOOTH TUBES

Figure 18 shows the variation of the normalized, local, condensing coefficient for five tubes. The data points lie about 40 percent above the curve predicted by the Nusselt theory. This close agreement was considered to be an indication of the proper opepation of the test apparatus and the data reduction procedures.

As discussed earlier in Chapter II, the Nusselt theory for a tube bundle is based on a number of basic assumptions. For example, in reality, the condensate does not fall as a continuous sheet as assumed for the Nusselt theory, but it forms drops before leaving the tube. This phenomenon has a tendency to result in a larger condensing coefficient than that predicted by the Nusselt theory.

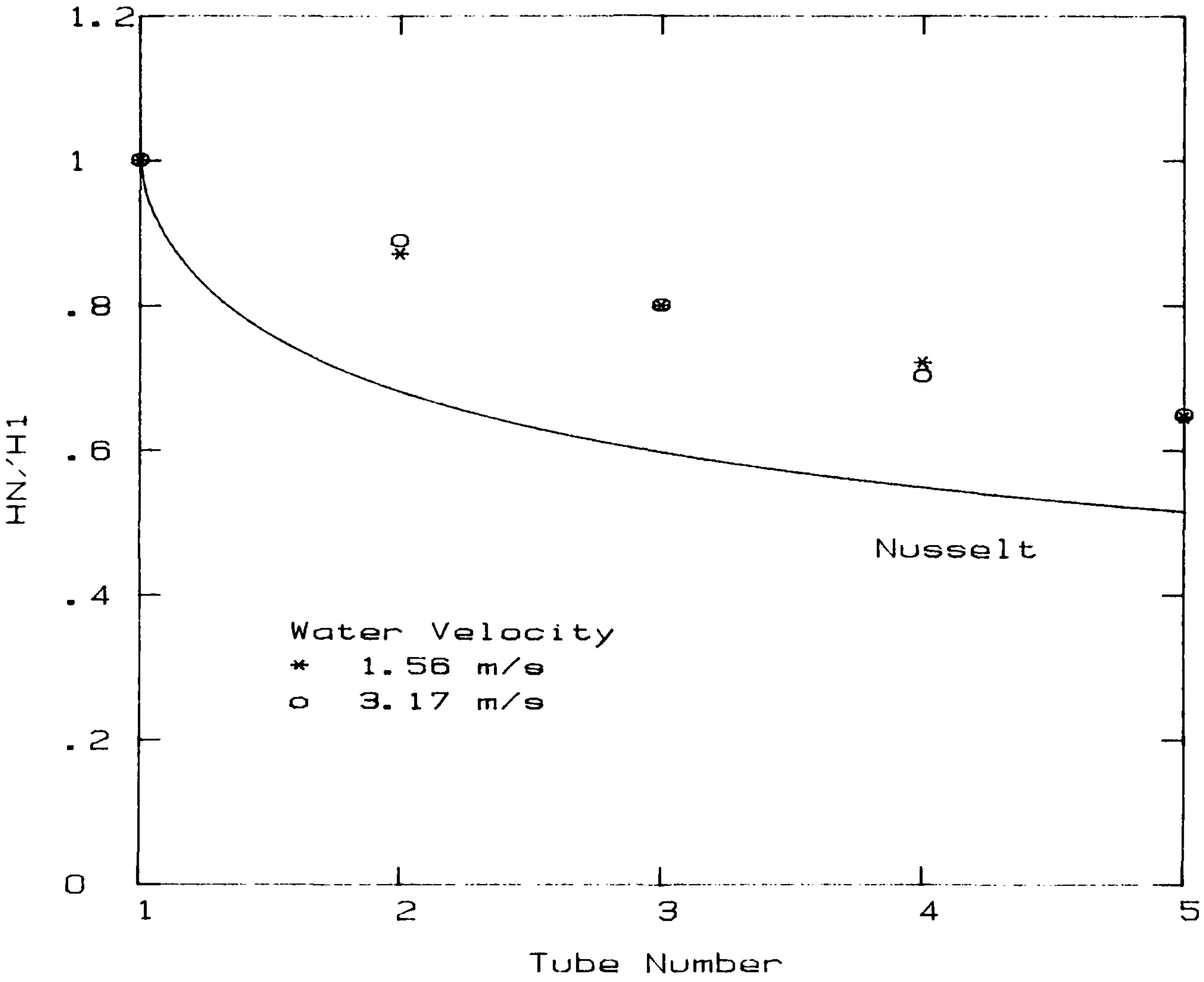


Figure 18. Variation of Local Condensing Coefficient with Tube Number (Run STNWNI-1)

Figure 19 shows the normalized, average condensing coefficient for five tubes. These data points show a much smoother trend than that for local values. Further, these data points are closer to the Eissenberg correlation than the Nusselt prediction.

Figure 20 shows the variation of the normalized, local, condensing coefficient for 30 tubes. The data points lie up to 35 percent above the curve predicted by the Nusselt theory. The outside heat-transfer coefficielt for the first tube was ten percent smaller than the value predicted by the Nusselt theory.

Figure 21 displays the normalized, average condensing coefficient for 30 tubes. Again, these data points show a smoother trend than that for local values. These points lie between the Eissenberg and Nusselt predictions.

Figure 22 shows the least-squares-fit curve for the data points for smooth tubes. The exponent derived is -0.154, and it is in close agreement with the value of -0.14 derived by Noftz [Ref. 12].

C. ROPED TUBES

Figure 23 displays the variation of the normalized, local, condensing coefficient for five tubes. The data points lie up to 15 percent above the curve representing the Nusselt theory.

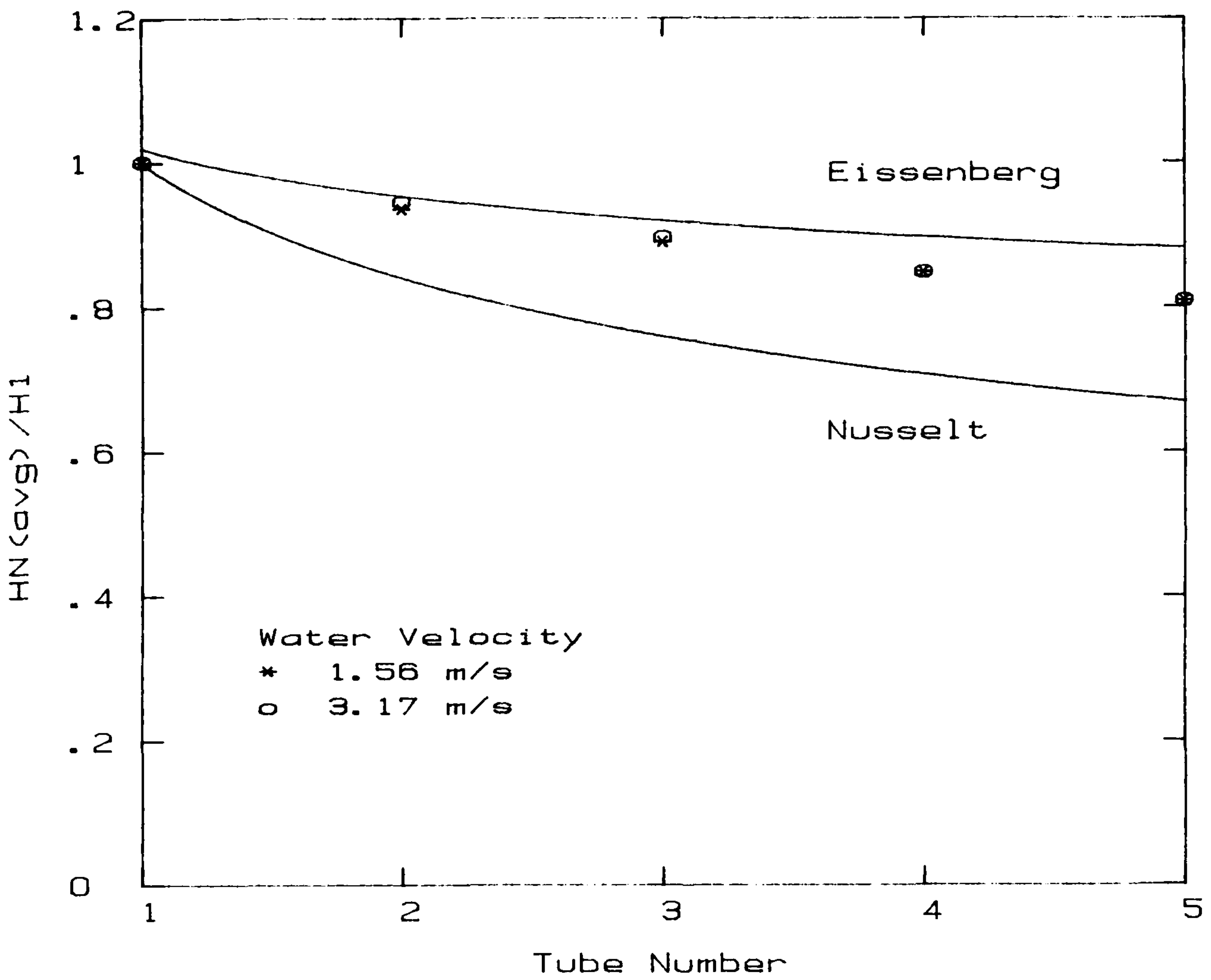


Figure 19. Variation of Average Condensing Coefficient with Tube Number (Run STNWNI-1)

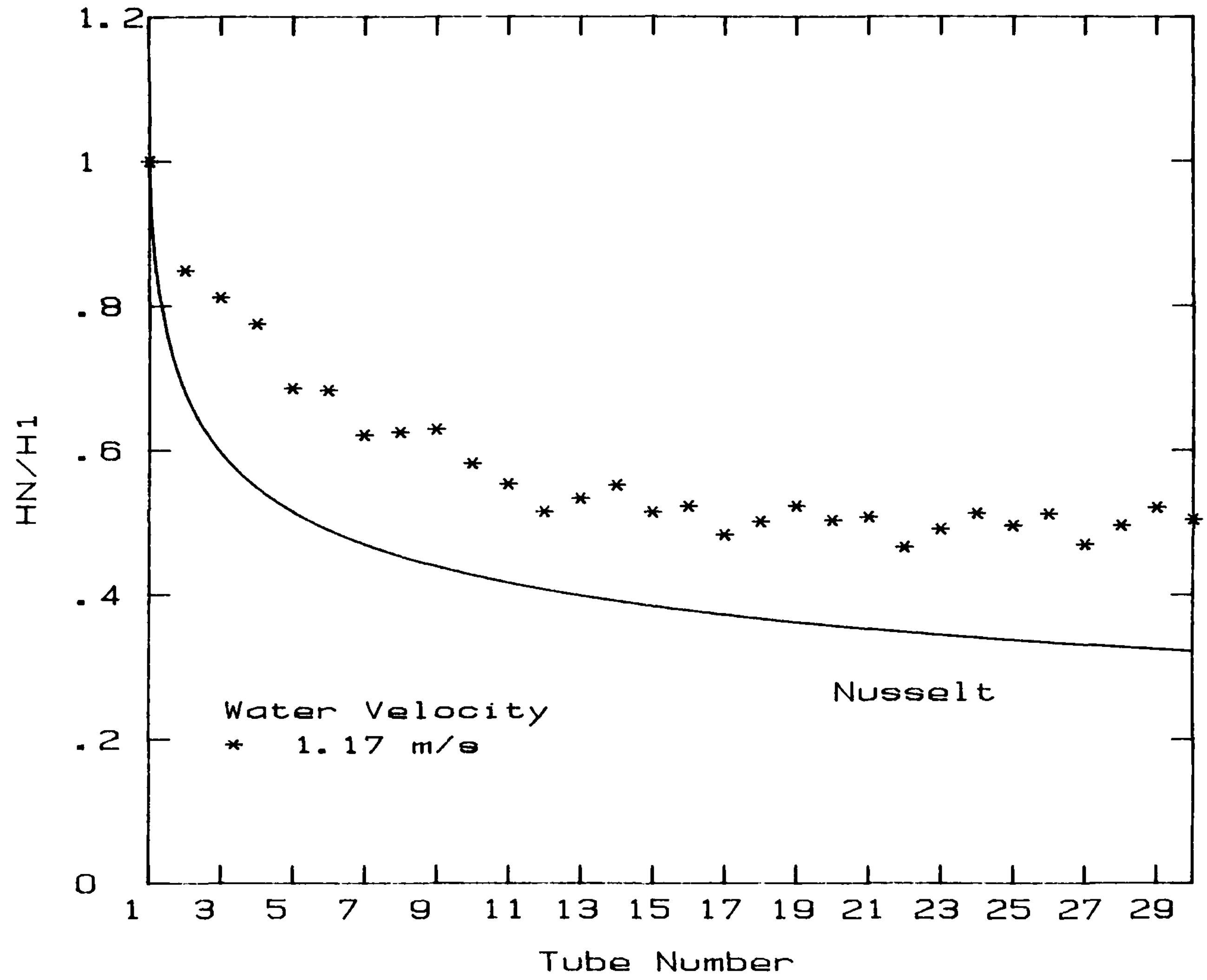


Figure 20. Variation of Local Condensing Coefficient with Tube Number (Run STNWI-1)

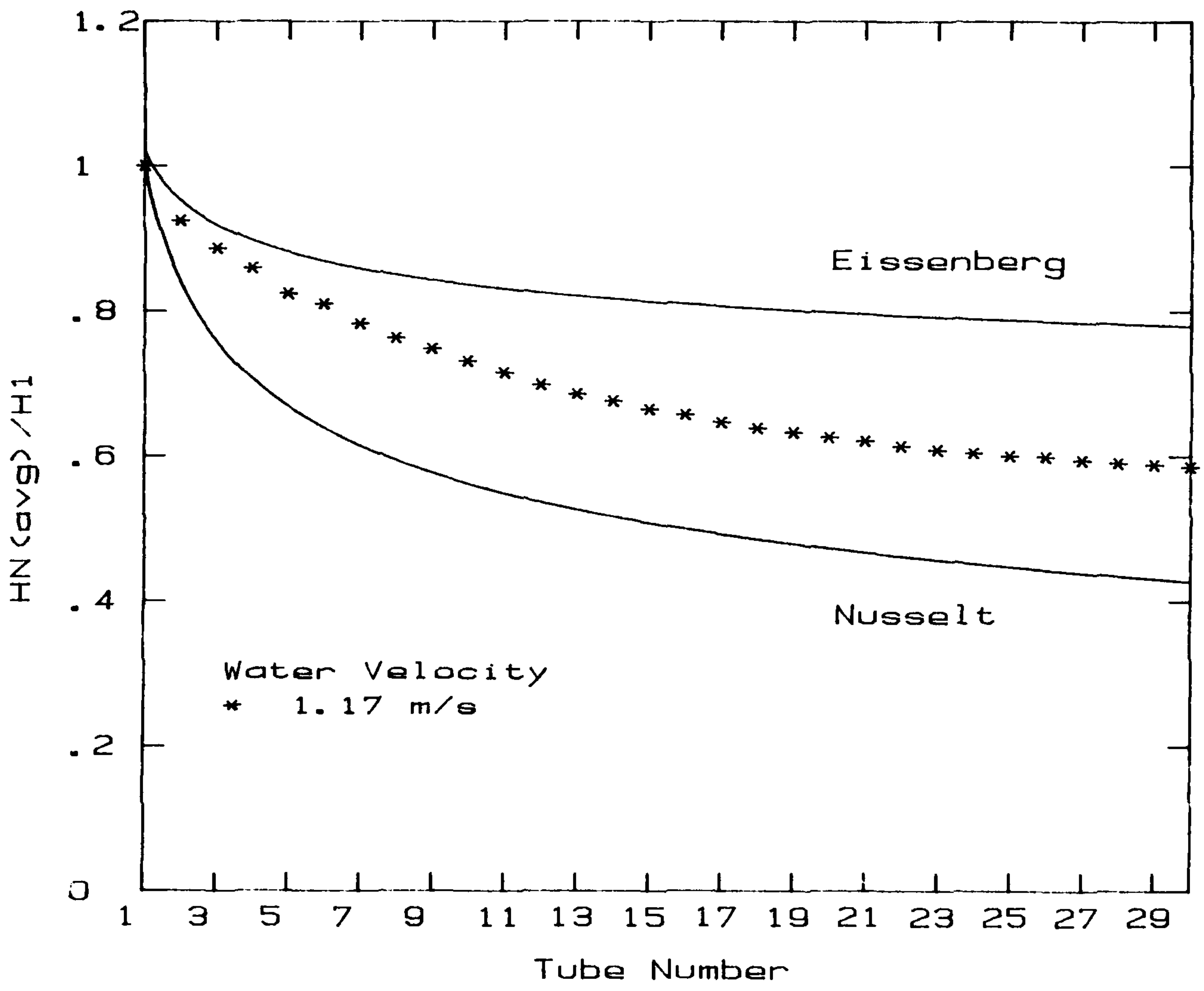


Figure 21. Variation of Average Condensing Coefficient with Tube Number (Run STNWI-1)

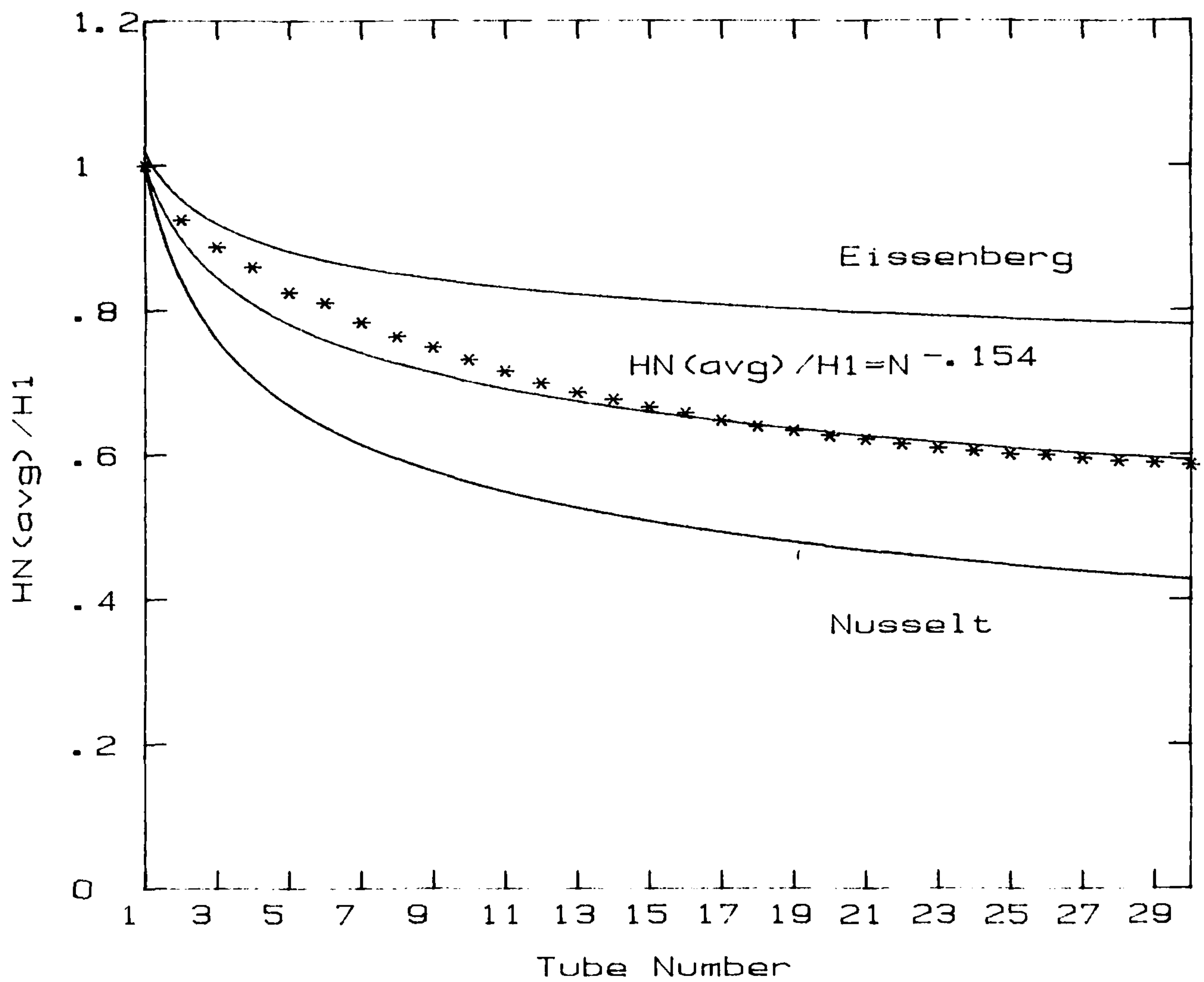


Figure 22. Least-Squares-Curve Fit for Data (Run STNW-1)

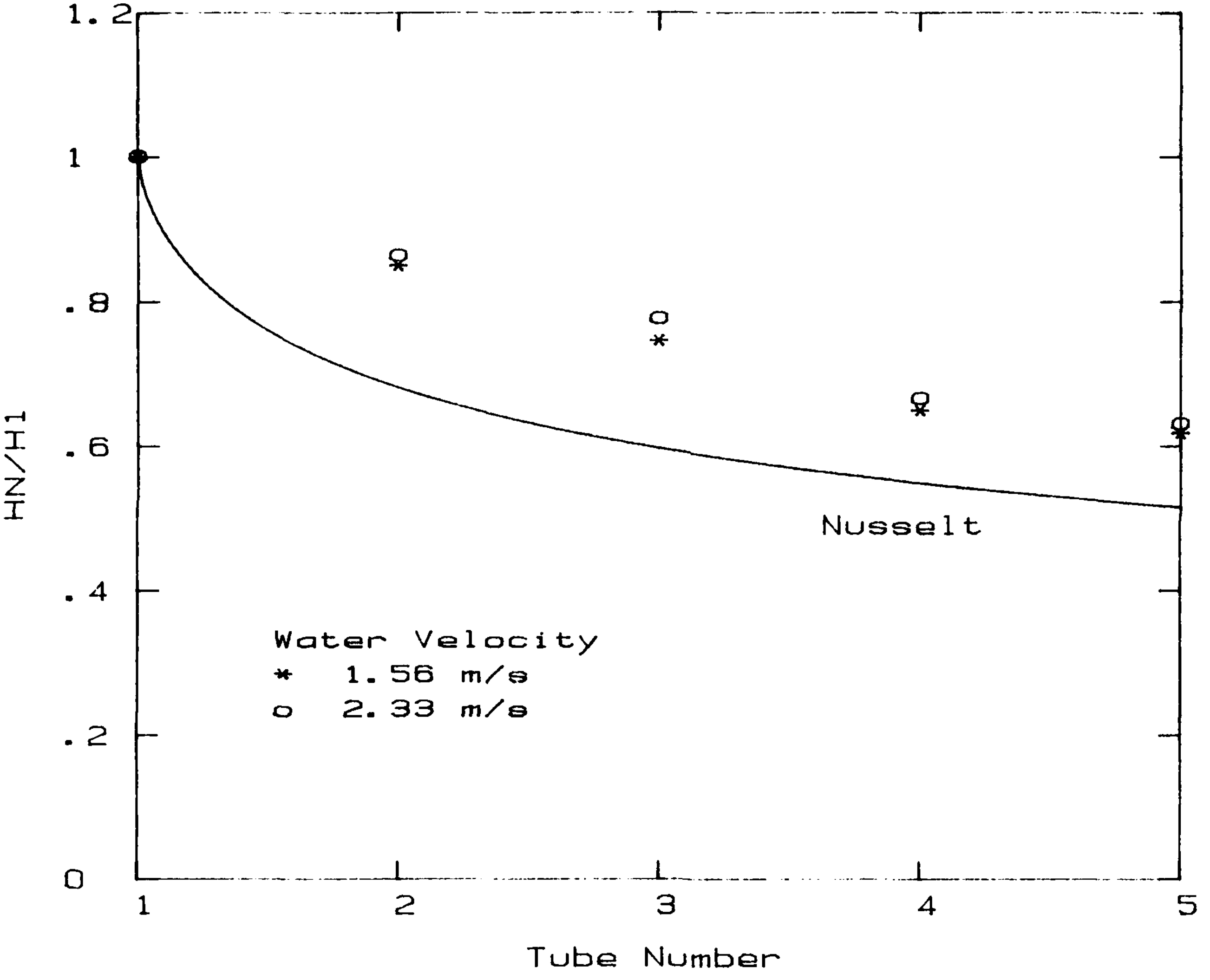


Figure 23. Variation of Local Condensing Coefficient with Tube Number (Run RTNWNI-3)

Figure 24 shows the normalized, average, condensing coefficient for five tubes. These points show a trend very close to the Eissenberg relationship.

Figure 25 represents the plot of the normalized, local, condensing coefficient under inundation conditions up to 30 tubes.

Figure 26 shows the normalized, average, condensing coefficient for 30 tubes. These data points lie about midway between the Eissenberg and Nusselt predictions.

Figure 27 shows the least-squares-fit curve for the data points for roped tubes. The exponent calculated is -0.183. Since the interest is in the large tube bundles. the curve fit was generated only for tubes 11 through 30. This is evident from the figure as the curve fit shows poor agreement when compared to the data points for the first ten tubes.

D. ROPED TUBES WRAPPED WITH WIRE

The outside. heat-transfer coefficient for the first tube was 11.3 percent greater than the value predicted by the Nusselt theory. This increase is in agreement with the manufacturer's claim that the special geometry of the roped tubes has a thinning effect on the condensate film, resulting in a larger condensing coefficient.

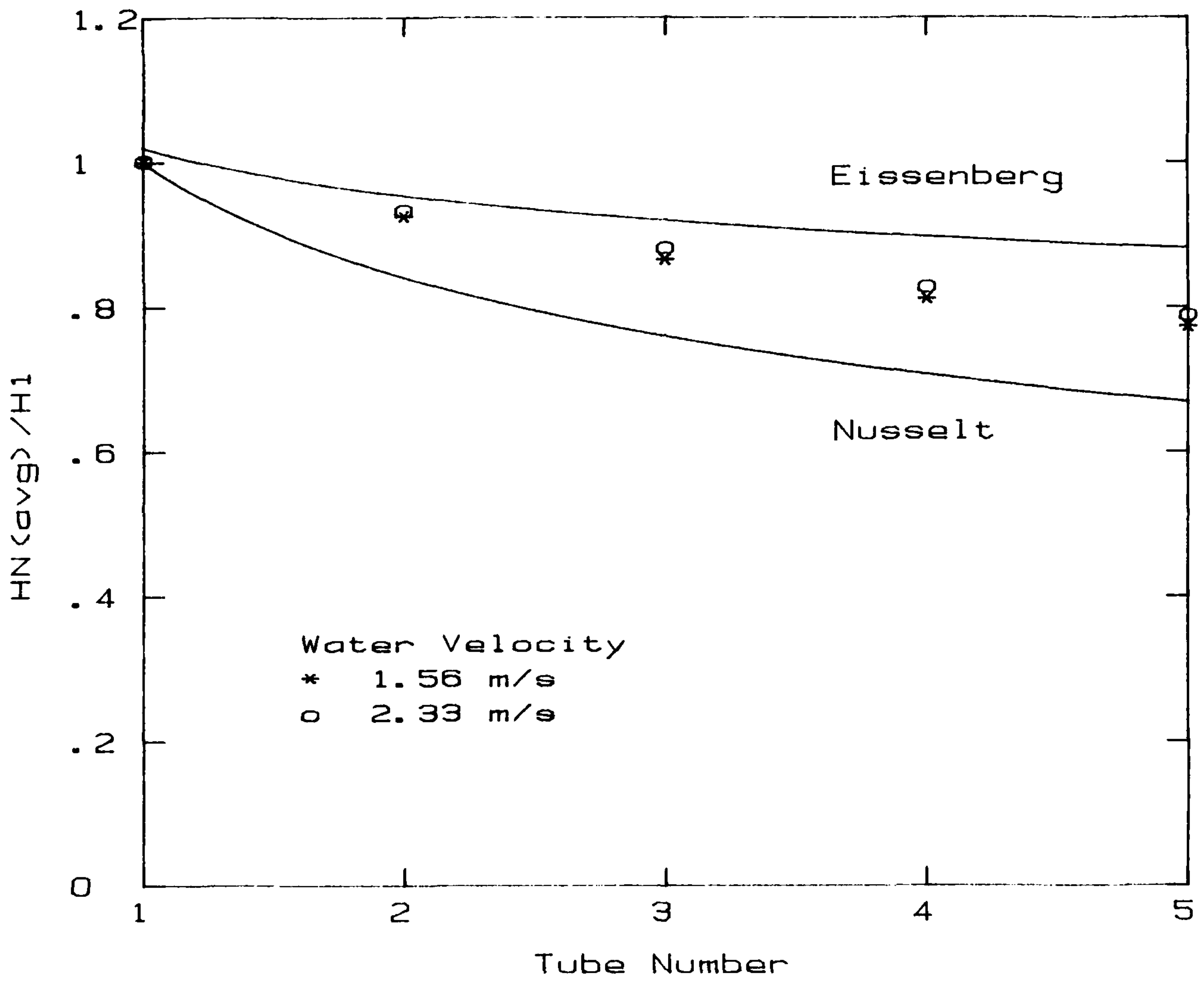


Figure 24. Variation of Average Condensing Coefficient with Tube Number (Run RTNWNI-3)

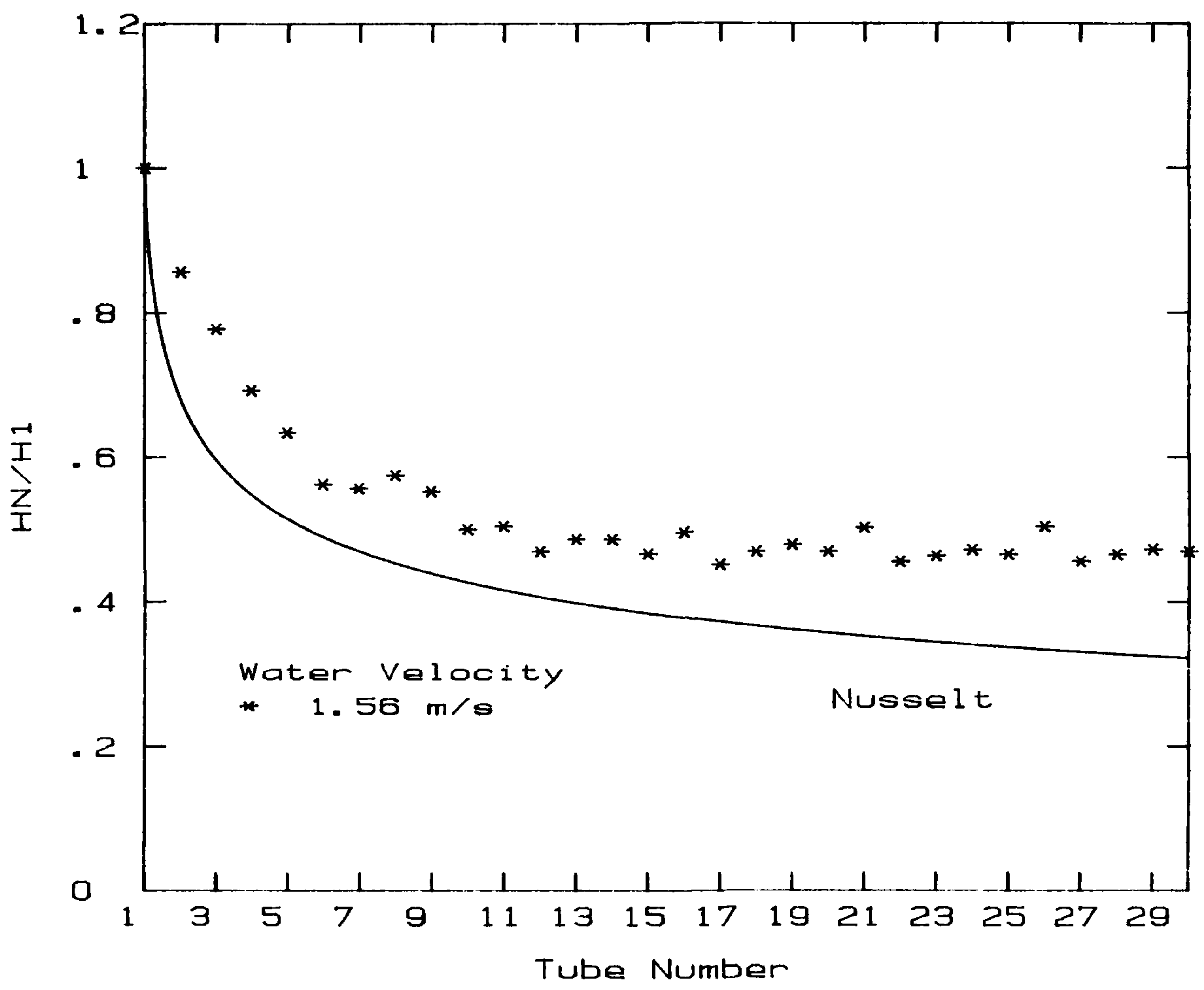


Figure 25. Variation of Local Condensing Coefficient with Tube Number (Run RTNWNI-3)

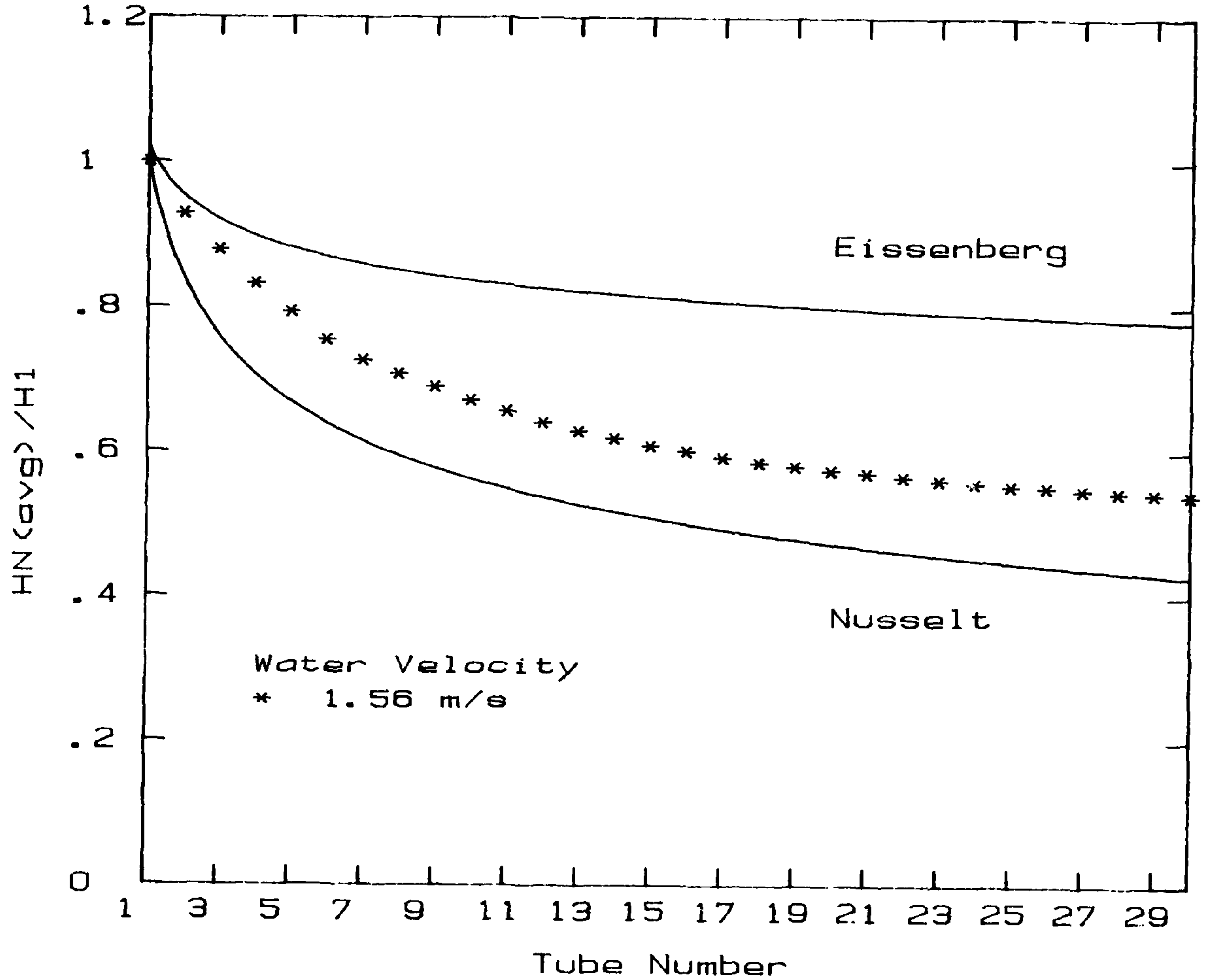


Figure 26. Variation of Average Condensing Coefficient with Tube Number (Run RTNWI-1)

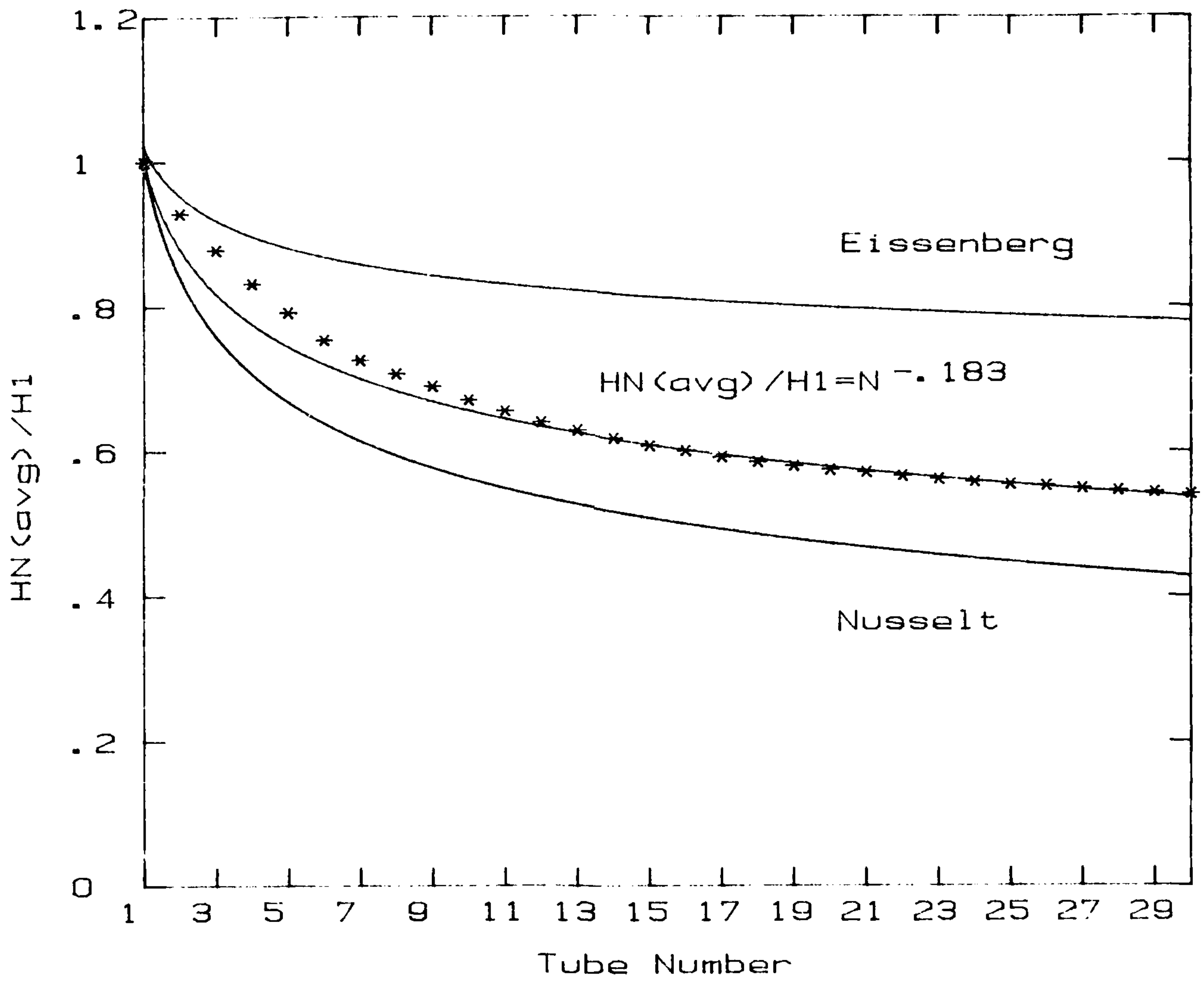


Figure 27. Least-Squares-Curve Fit for Data (Run RTNWI-1)

Figure 28 shows the variation of the normalized, local, condensing coefficient for five tubes. The data points lie up to 63 percent above the curve predicted by the Nusselt theory.

Figure 29 displays the normalized, average, condensing coefficient. The data points are well above the Eissenberg correlation.

Figure 30 shows the normalized, local. condensing coefficient for a bundle of 30 tubes. The data points are scattered within the limits of uncertainty, as predicted by the error analysis (see Appendix C).

Fiqure 31 shows the variation of the data points representing the normalized, average, condensinq coefficient under inundation up to 30 tubes. The data points lie up to 100 percent above the curve predicted by the Nusselt theory.

Figure 32 represents the least-squares-fit curve, which has a derived exponent of -0.039. The curve fit is in good agreement with all the data points.

E. SMOOTH TUBES WRAPPED WITH WIRE

The outside heat-transfer coefficient for the first tube was up to six percent greater than the value predicted by the Nusselt theory. This increase is caused by the thinning effect on the condensate film, resulting from the surface-tension forces acting toward the wrapped wire.

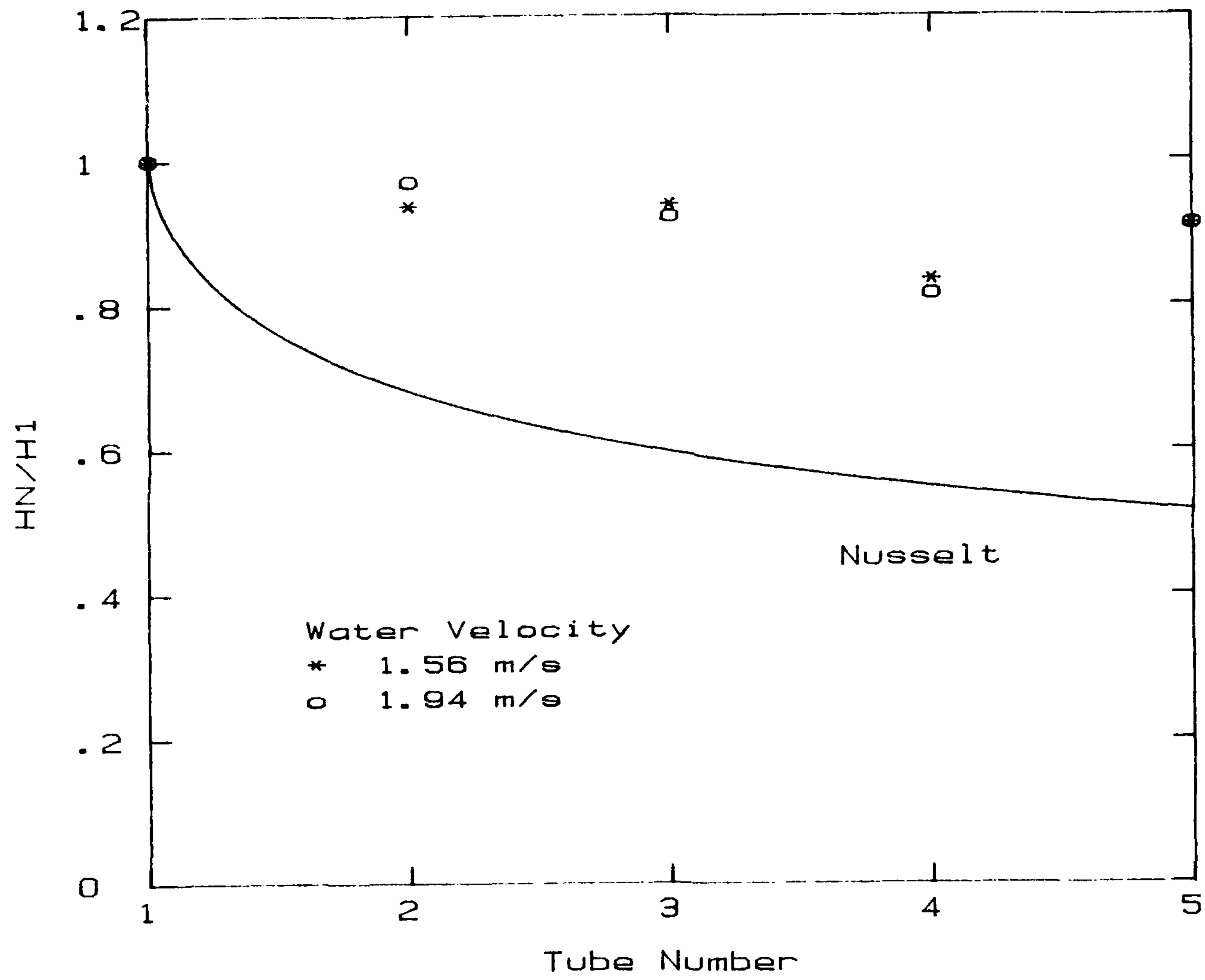


Figure 28. Variation of Local Condensing Coefficient with Tube Number (Run RTWNI-3)

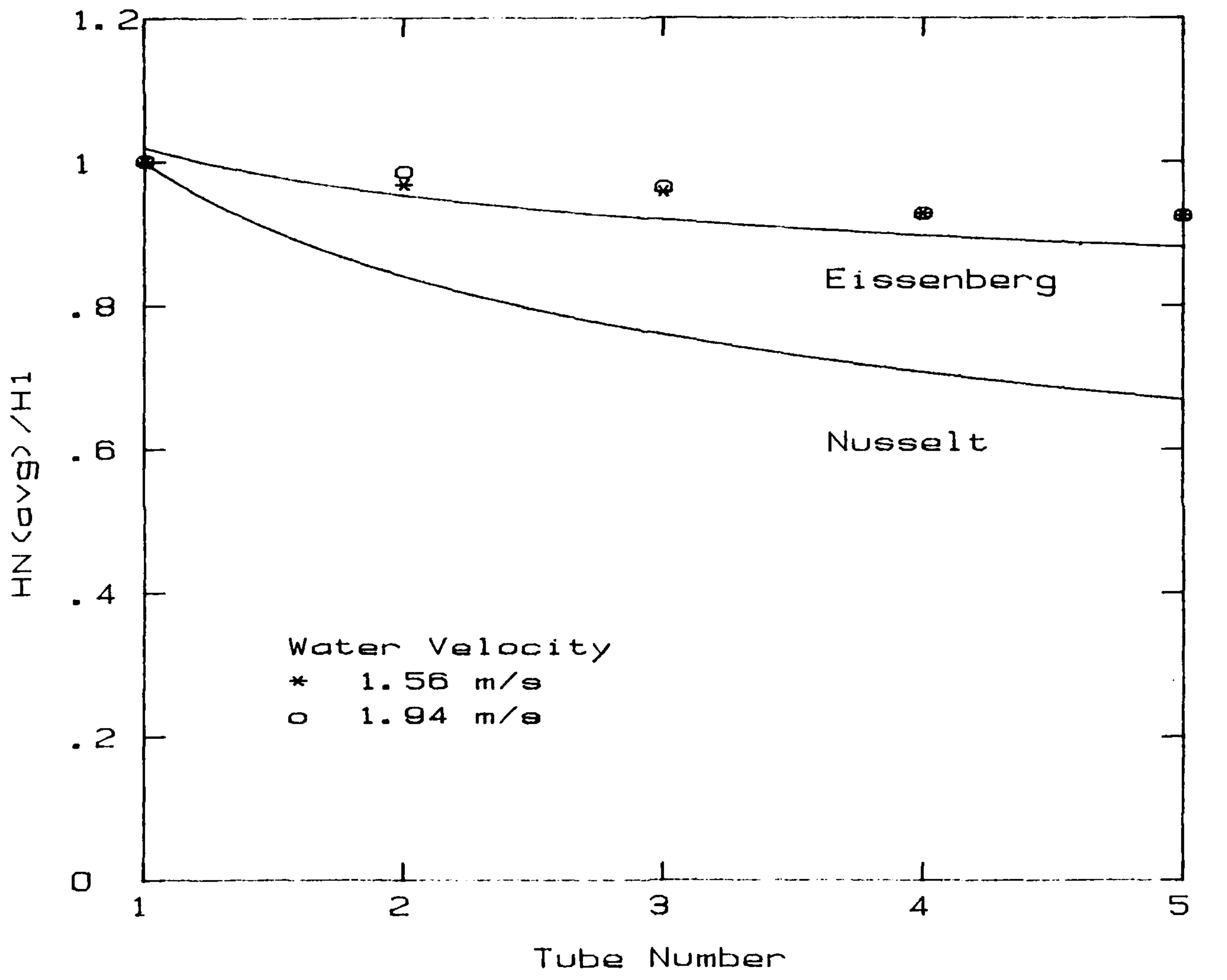


Figure 29. Variation of Average Condensing Coefficient with Tube Number (Run RTWNI-3)

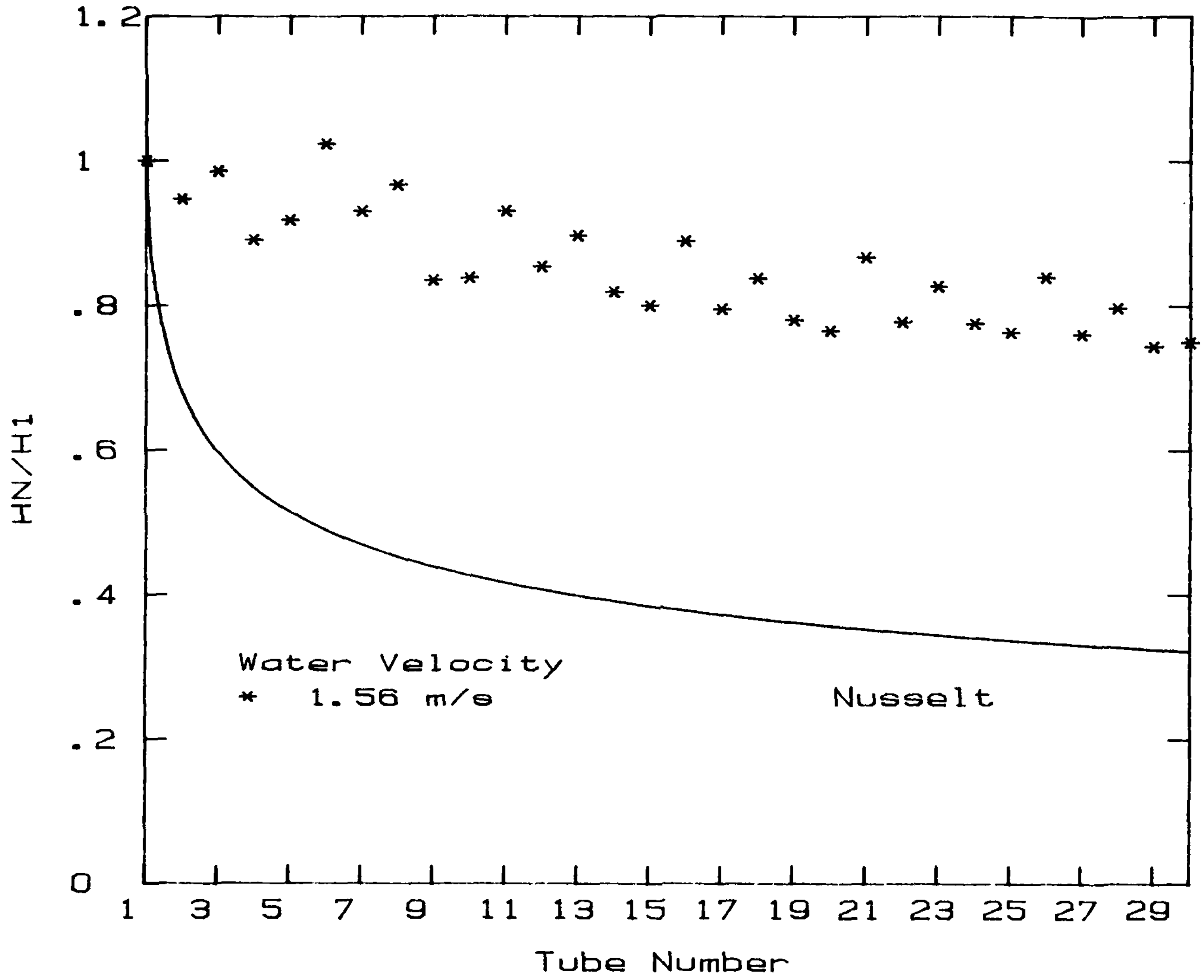


Figure 30. Variation of Local Condensing Coefficient with Tube Number (Run RTWI-3)

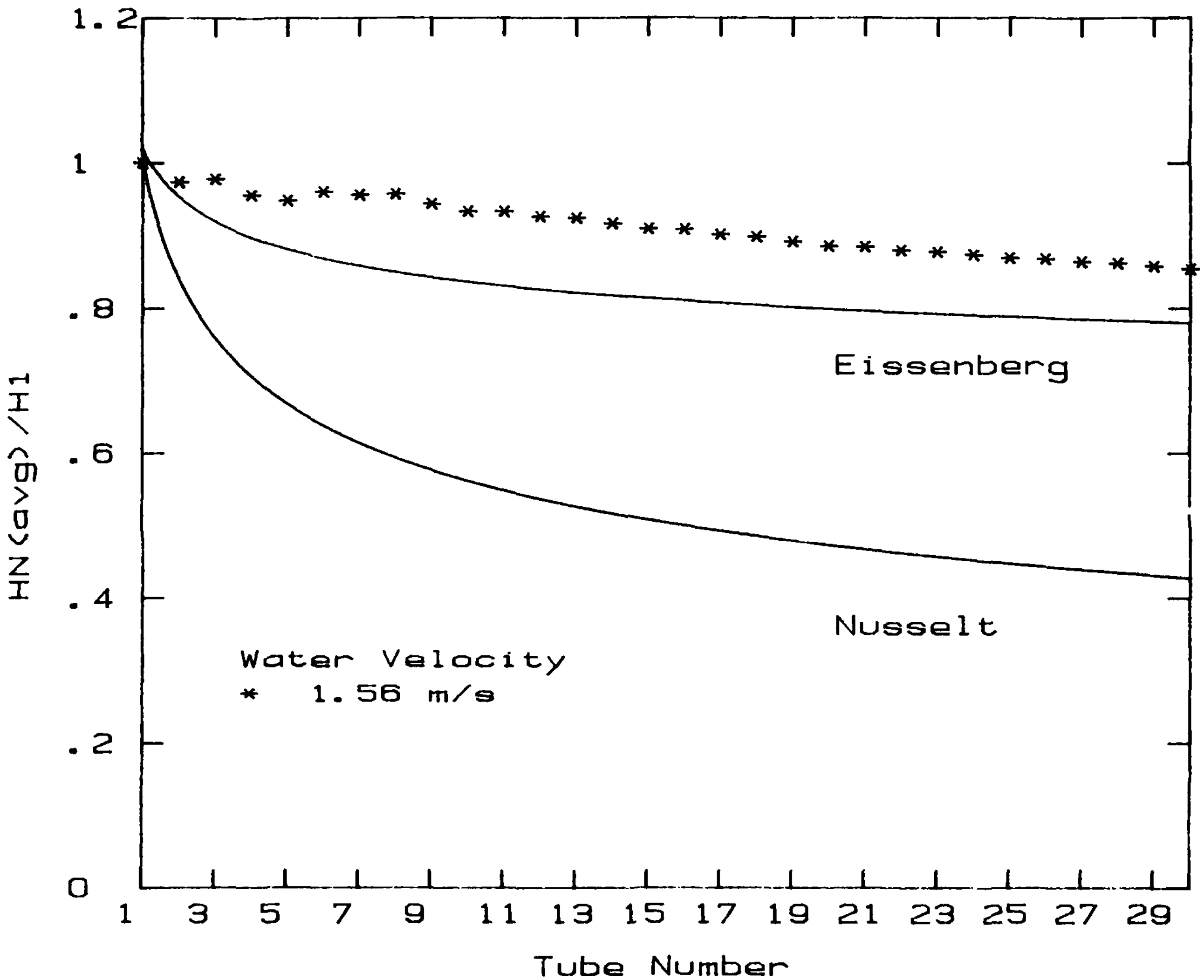


Figure 31. Variation of Average Condensing Coefficient with Tube Number (Run RTWI-1)

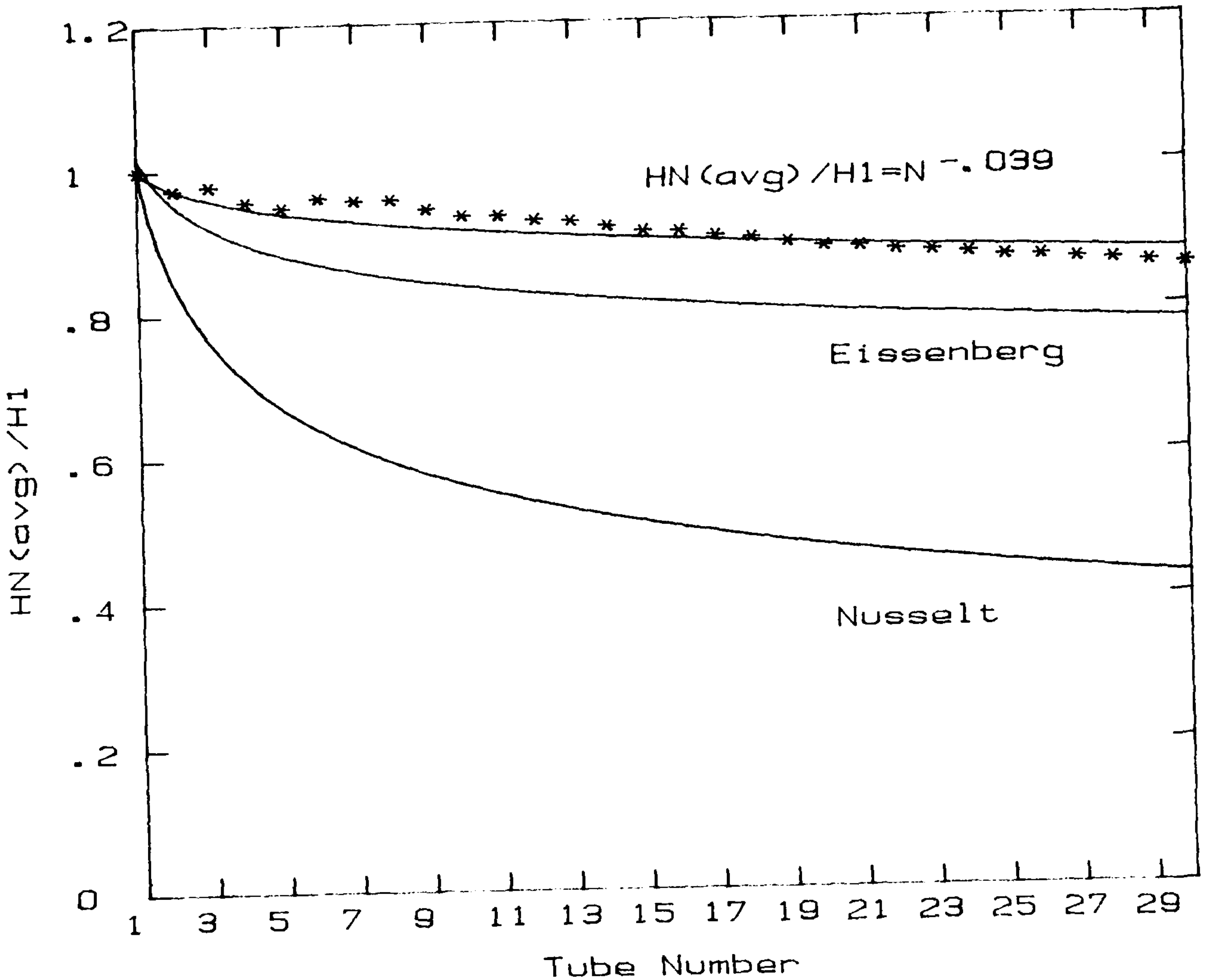


Figure 32. Least-Squares-Curve Fit for Data (Run RTWI-1)

Figure 33 shows the variation of the normalized, local, condensing coefficient for five tubes. The data points lie up to 76 percent above the curve predicted by the Nusselt theory.

Figure 34 shows the normalized, average condensing coefficient for five tubes. These data points are well above the Eissenberg correlation.

Figure 35 shows the variation of the normalized, local, condensing coefficient for 30 tubes. The data points lie up to 107 percent above the curve predicted by the Nusselt theory.

Figure 36 displays the normalized, average. condensing coefficient for 30 tubes. The data points lie above the curve representing the Eissenberg correlation.

Figure 37 shows that the curve fit is in good agreement with all the data points. The derived exponent is -0.037.

F. OBSERVATIONS

1. Smooth Tubes

During all runs, complete film-wise condensation was observed without any visible evidence of drop-wise condensation. This was mainly achieved by the tube cleaning procedures used.

A slow rate of condensate droplet migration. from cooling water outlet end to the inlet end, was observed at the bottom of each active tube. This problem was minimized

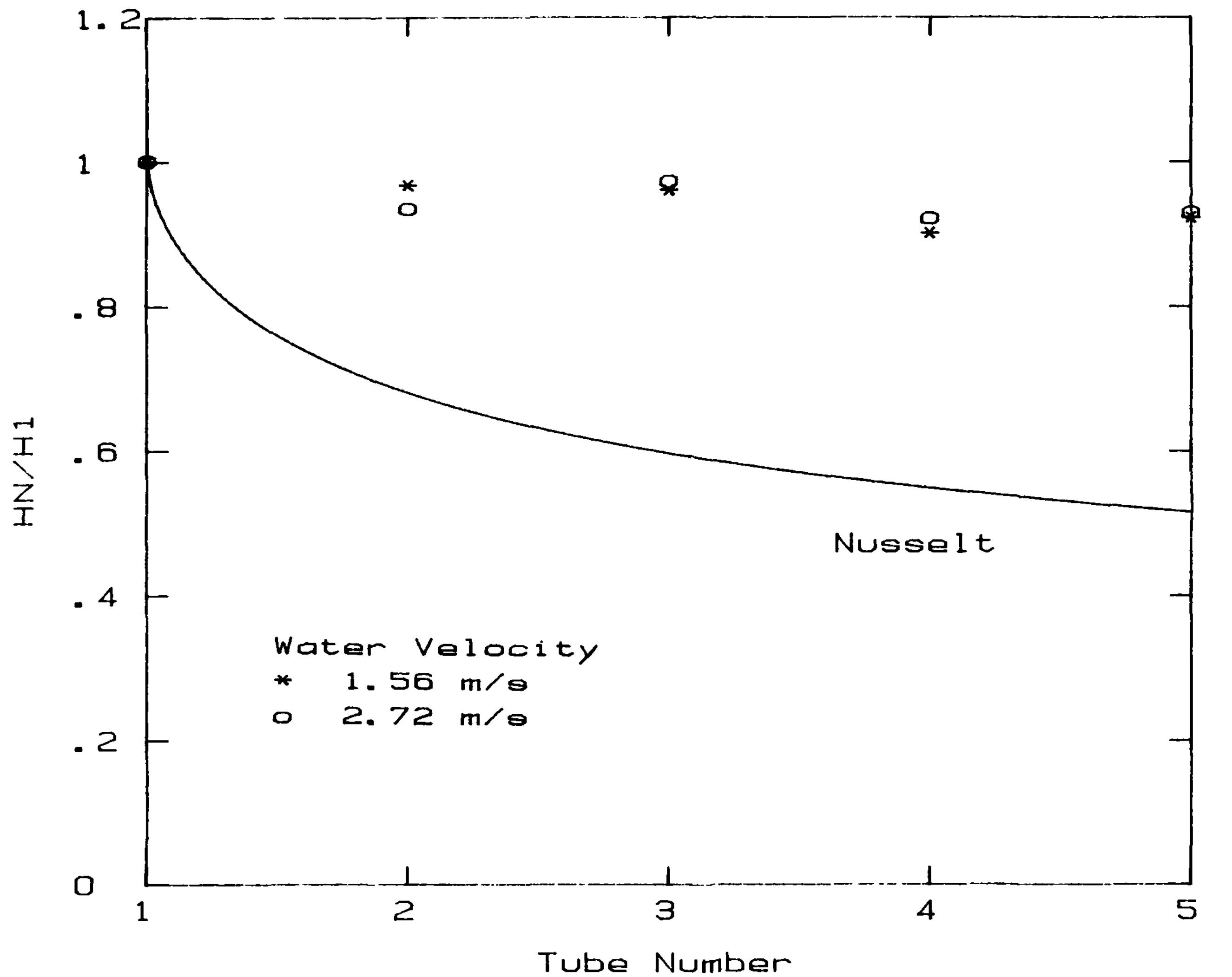


Figure 33. Variation of Local Condensing Coefficient with Tube Number (Run STWNI-2)

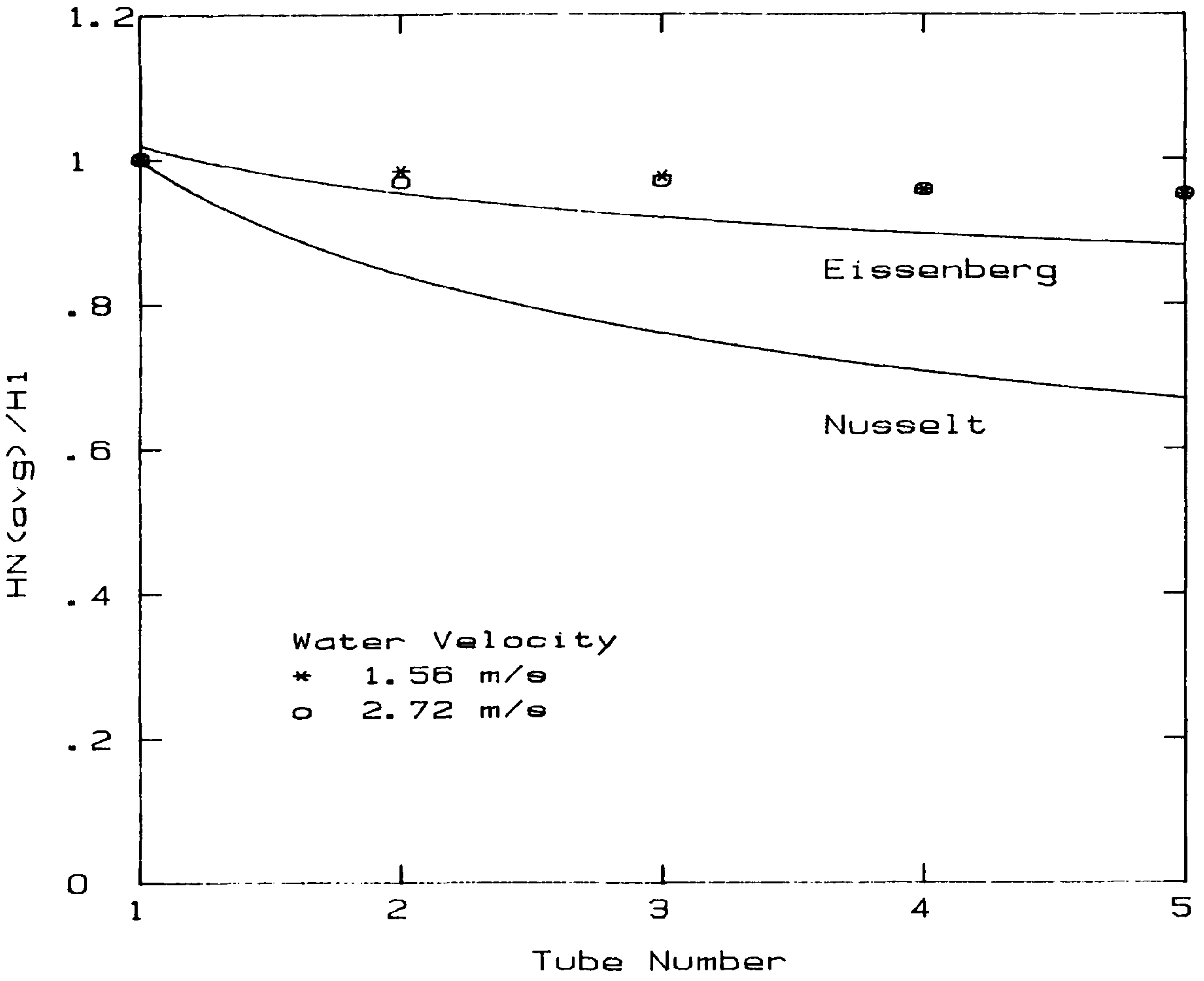


Figure 34. Variation of Average Condensing Coefficient with Tube Number (Run STWNI-2)

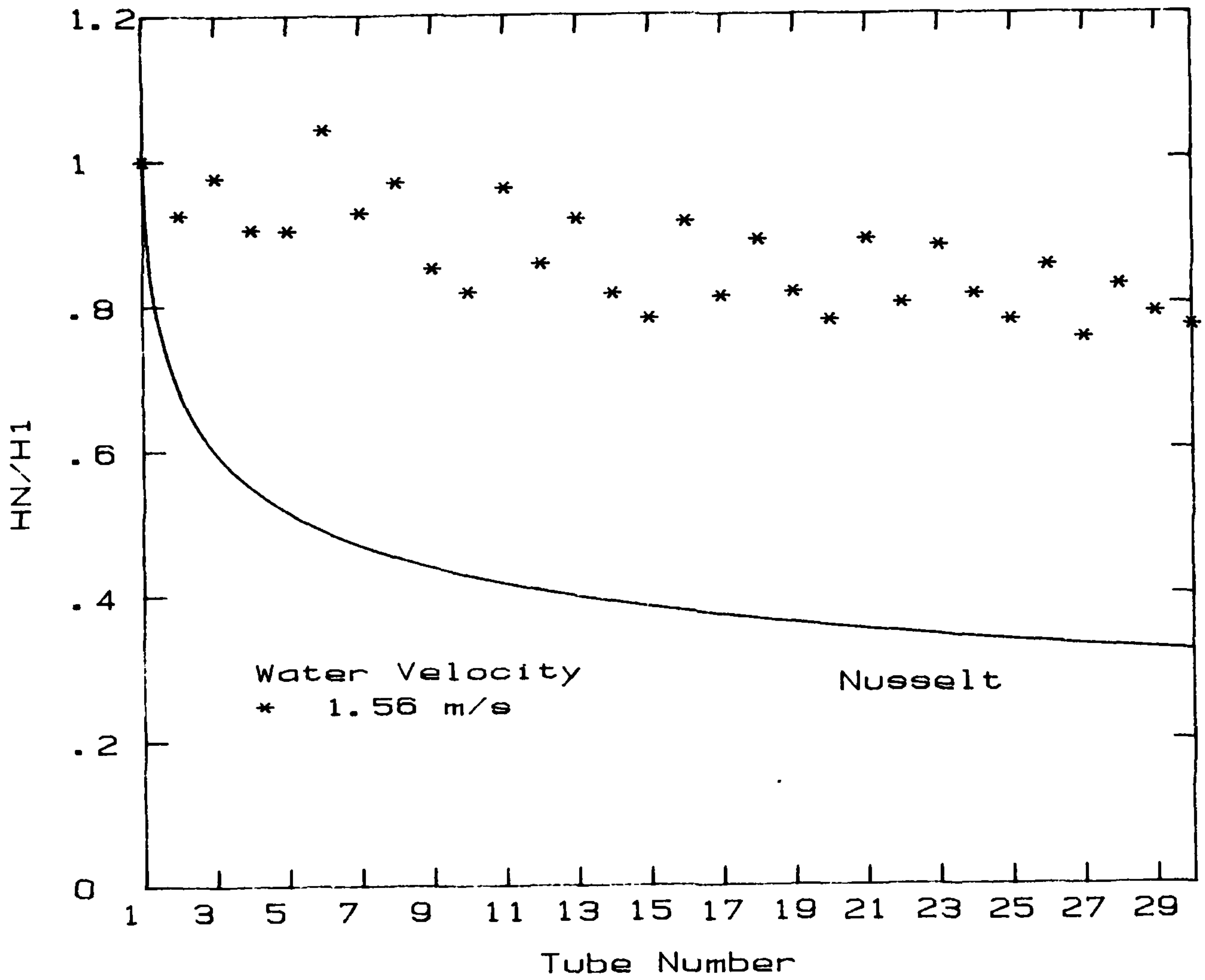


Figure 35. Variation of Local Condensing Coefficient with Tube Number (Run STWI-1)

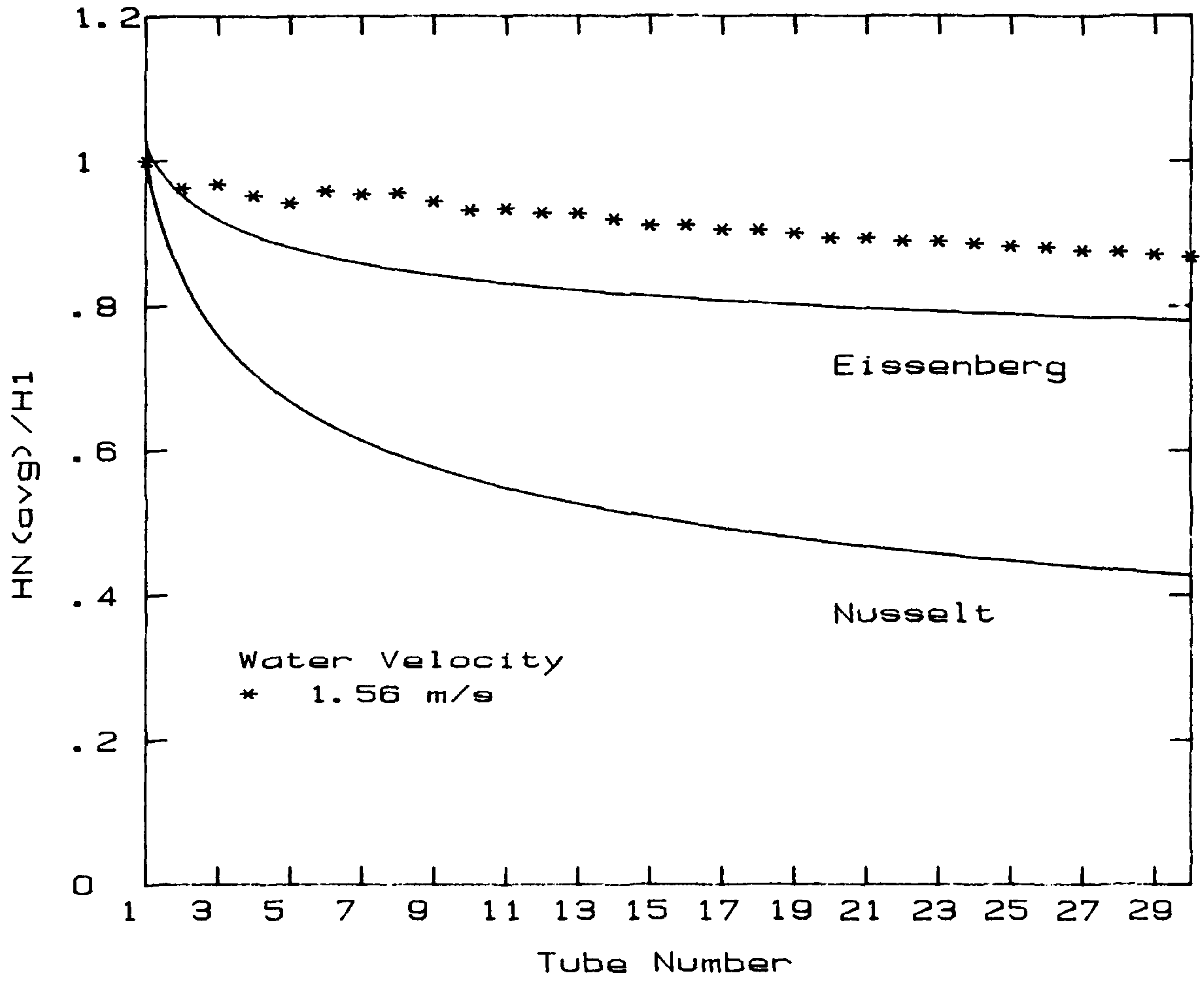


Figure 36. Variation of Average Condensing Coefficient with Tube Number (Run STWI-1)

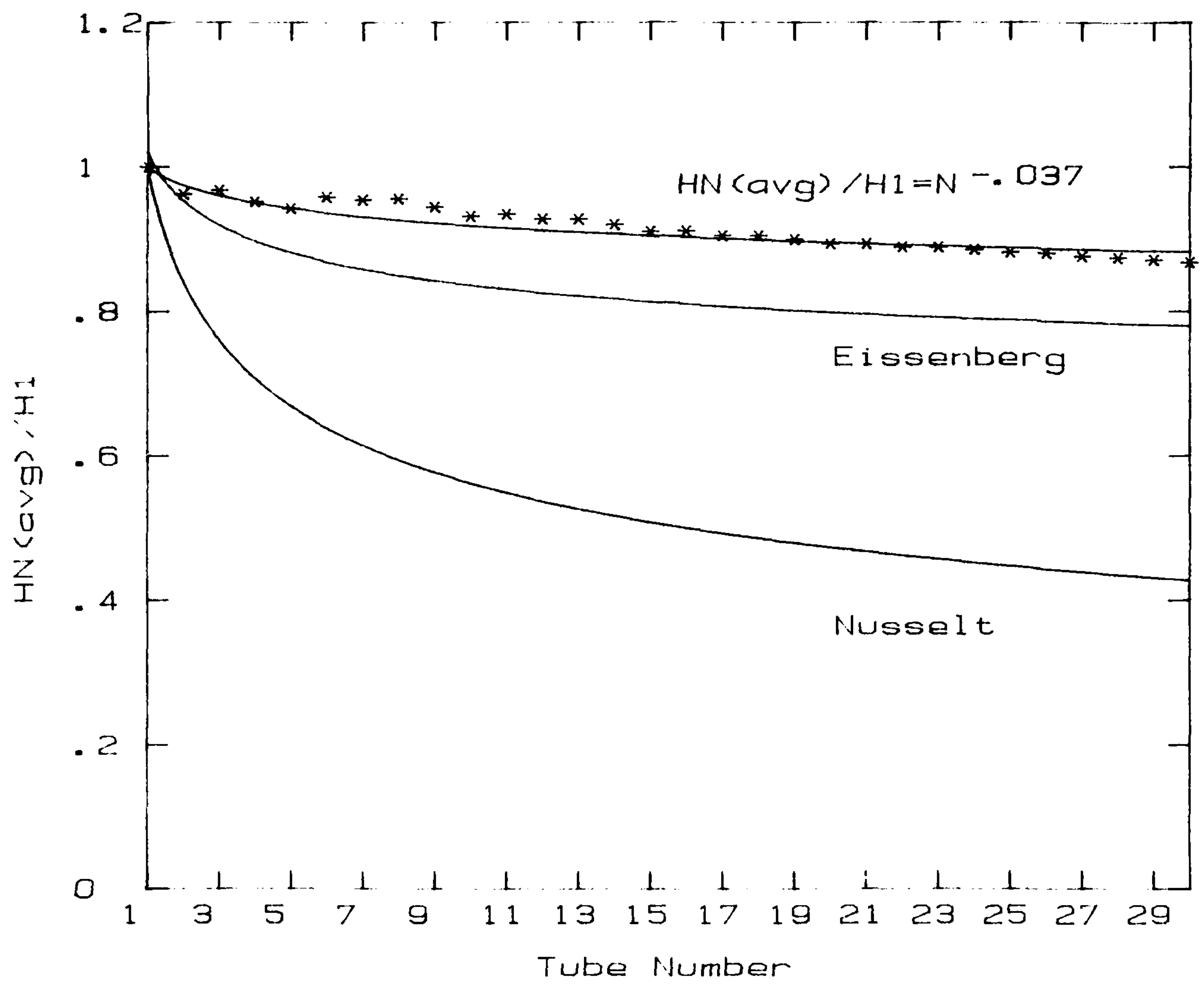


Figure 37. Least-Squares-Curve Fit for Data (Run STWI-1)

TABLE IV

SUMMARY OF RUNS WITH INUNDATION

File Name	Tube Type	External Wire Wrapped or Not
STNWI-1	Smooth	No
RTNWI-1	Roped	No
RTNWI-2	Roped	No
RTNWI-3	Roped	No
RTNWI-4	Roped	No
RTNWI-5	Roped	No
RTWI-1	Roped	Yes
RTWI-2	Roped	Yes
STWI-1	Smooth	Yes
STWI-2	Smooth	Yes

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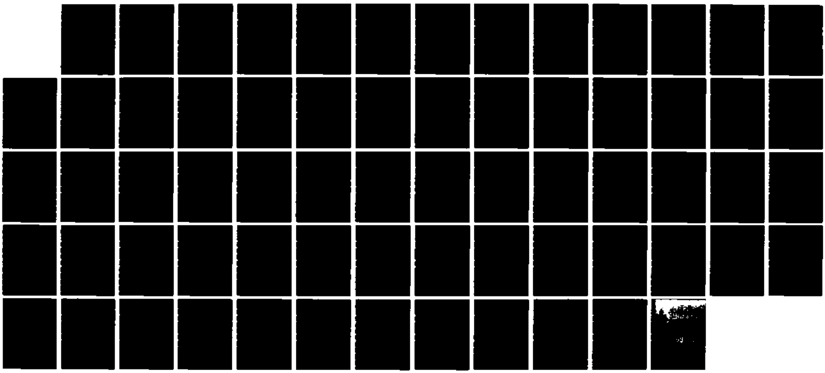

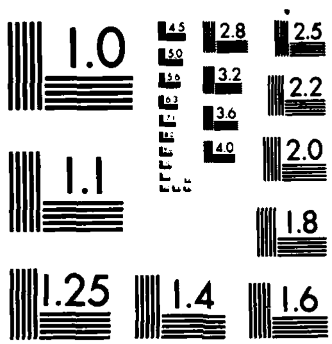

MICROCOPY RESOLUTION TEST CHART
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TABLE V

SUMMARY OF RUNS WITHOUT INUNDATION

File Name	Tube Type	External Wire Wrapped or Not
STNWEV-2	Smooth	No
STNWNI-1	Smooth	No
STNWNI-2	Smooth	No
STNWNI-3	Smooth	No
RTNWNI-3	Roped	No
RTWNI-1	Roped	Yes
RTWNI-2	Roped	Yes
RTWNI-3	Roped	Yes
STWNI-2	Smooth	Yes
STWNI-4	Smooth	Yes
STWNI-5	Smooth	Yes

TABLE VI

RESULTS FOR RUNS WITHOUT INUNDATION

File Name	Tube Type	External Wire Wrap or not	h_N/h_1	$\bar{h}_N/h_1$	Cooling Water Velocity (m/s)
STNWNI-1	Smooth	No	0.7476	0.8662	1.56
-1	Smooth	No	0.7586	0.8722	3.17
STNWNI-2	Smooth	No	0.6522	0.8031	1.56
-2	Smooth	No	0.6503	0.8102	3.17
STNWNI-3	Smooth	No	0.6473	0.8031	1.56
-3	Smooth	No	0.6393	0.8002	3.17
RTNWNI-3	Roped	No	0.6181	0.7728	1.56
-3	Roped	No	0.6298	0.7866	2.33
RTWNI-1	Roped	Yes	0.9465	0.9466	1.56
RTWNI-2	Roped	Yes	0.9479	0.9579	1.56
-2	Roped	Yes	0.9250	0.9303	2.33
RTWNI-3	Roped	Yes	0.9098	0.9244	1.56
-3	Roped	Yes	0.9094	0.9228	1.94
STWNI-2	Smooth	Yes	0.9278	0.9508	1.56
-2	Smooth	Yes	0.9224	0.9509	2.72

TABLE VI (continued)

STWNI-4	Smooth	Yes	0.8404	0.9051	1.56
-4	Smooth	Yes	0.8496	0.8713	2.72
STWNI-5	Smooth	Yes	0.9117	0.9545	1.56
-5	Smooth	Yes	0.9322	0.9506	2.72

TABLE VII

RESULTS FOR RUNS WITH INUNDATION UP TO 30 TUBES

File Name	Tube Type	External Wire Wrap or not	h_N/h_1	$\bar{h}_N/h_1$	Cooling Water Velocity (m/s)
STNWI-1	Smooth	No	0.504	0.5857	1.17
RTNWI-1	Roped	No	0.4694	0.5404	1.56
RTNWI-2	Roped	No	0.4751	0.5306	2.72
RTNWI-3	Roped	No	0.5039	0.5554	1.56
RTNWI-4	Roped	No	0.4885	0.5420	1.56
RTNWI-5	Roped	No	0.5029	0.5543	2.72
STWI-1	Smooth	Yes	0.7693	0.8674	1.56
STWI-2	Smooth	Yes	0.7227	0.8309	1.56
RTWI-1	Smooth	Yes	0.75	0.8536	1.56
RTWI-2	Smooth	Yes	0.7876	0.8623	1.56

TABLE VIII

COMPARISON OF HEAT-TRANSFER COEFFICIENTS FOR TUBE #1 IN THE BUNDLE (WITHOUT INUNDATION) AT 1.56 m/s COOLING WATER VELOCITY

File Name	Tube Type	External Wire Wrap or not	h_1 (W/m^2k)	h_{Nu} (W/m^2k)
STNWNI-1	Smooth	No	11136.4	10801.9
			10257.7	11023.9
			10448.9	10991.6
			10328.9	11018.6
			9960.9	11102.2
STNWNI-2	Smooth	No	10521.7	10915.7
			10624.5	10902.4
			10690.9	10893.9
			10799.4	10879.2
			10597.9	10909.1
STNWNI-3	Smooth	No	10453.1	11050.4
			10620.3	11028.9
			10643.8	11027.8
			10613.8	11032.3
			10628.7	11027.9
RTNWNI-3	Roped	No	11329.	9758.6
			11423.3	9747.2
			11328.7	9757.5
			11359.2	9755.1
			11470.4	9744.6
RTWNI-1	Roped	Yes	9946.4	9989.1
			9975.8	9908.5
			9889.7	9908.5
			10052.3	9873.
			10019.1	9884.7

TABLE VIII (continued)

RTWNI-2	Roped	Yes	9951.3	9946.5
			9933.6	9960.4
			9954.1	9953.5
			9832.2	9973.3
			9931.1	9960.6
RTWNI-3	Roped	Yes	10594.4	9511.4
			10188.2	9612.6
			10040.8	9648.7
			10104.	9645.9
			10328.9	9607.
STWNI-2	Smooth	Yes	10624.7	11056.5
			10681.8	11051.
			10746.1	11047.5
			10772.1	11044.9
			10789.3	11044.7
STWNI-4	Smooth	Yes	11846.2	10809.
			11275.	10878.9
			11259.3	10882.1
			11299.5	10878.6
			11285.8	10887.
STWNI-5	Smooth	Yes	10607.6	10994.6
			10542.9	11010.1
			10854.4	10962.7
			10681.3	10991.2
			10698.3	10984.9

TABLE IX

COMPARISON OF HEAT-TRANSFER COEFFICIENT FOR TUBE WITH INUNDATION

File Name	Tube Type	External Wire Wrap or not	h_i ($^W/_{m^2k}$)	h_{Nu} ($^W/_{m^2k}$)
STNWI-1	Smooth	No	11005.6	12102.7
			11132.5	12089.5
			11219.7	12079.5
			11126.1	12086.2
			11028.4	12100.1
RTNWI-1	Roped	No	11359.8	9785.8
			11394.3	9783.7
			11461.4	9777.1
			11329.1	9806.1
			11351.3	9796.1
RTNWI-2	Roped	No	11089.7	9225.9
			11103.1	9221.
			11096.7	9225.6
			11058.4	9237.6
			10865.6	9281.3
RTNWI-3	Roped	No	10790.5	9793.2
			11056.1	9741.9
			10932.4	9764.3
			11004.2	9743.3
			11030.8	9734.1
RTNWI-4	Roped	No	11999.5	9502.2
			11322.1	9624.6
			11329.9	9622.3
			11239.5	9631.4
			11389.2	9616.5

TABLE IX (continued)

RTNWI-5	Roped	No	10493.5	9312.8
			10657.8	9285.9
			10894.	9241.1
			10663.6	9297.8
			10635.5	9313.8
RTWI-5	Roped	Yes	9909.75	9972.6
			9881.61	9980.3
			9956.31	9964.8
			9862.91	9981.2
			9923.13	9970.5
RTWI-2	Roped	Yes	9953.69	9837.7
			10026.1	9830.7
			10003.7	9844.9
			10085.7	9826.0
			10095.9	9826.1
STWI-1	Roped	Yes	10777.9	11048.4
			10785.3	11046.4
			10800.3	11048.4
			10767.1	11050.2
			10754.7	11051.
STWI-2	Roped	Yes	11461.	10909.8
			11731.7	10869.0
			11533.1	10897.8
			11440.1	10908.0
			11561.0	10891.1

TABLE X

COMPARISON OF $\bar{h}_N/h_{Nu}$ FOR UNINUNDATED TUBE RUNS

Tube Type	External Wire Wrap or not	$\bar{h}_N/h_{Nu}$
Smooth	No	0.7986
Roped	No	0.9019
Roped	Yes	0.9651
Smooth	Yes	0.9339

TABLE XI

COMPARISON OF h_N/h_{Nu} FOR INUNDATION TUBE RUNS

Tube Type	External Wire Wrap or not	$\bar{h}_N/h_{Nu}$
Smooth	No	0.59
Roped	No	0.63
Roped	Yes	0.86
Smooth	Yes	0.86

TABLE XII

EXPONENTS OF THE LEAST-SQUARES-FIT

File Name	Tube Type	External Wire Wrap or not	Exponent
STNWI-1	Smooth	No	0.154
RTNWI-1	Roped	No	0.183
-2	Roped	No	0.191
-3	Roped	No	0.179
-4	Roped	No	0.185
RTWI-1	Roped	Yes	0.039
-2	Roped	Yes	0.039
STWI-1	Smooth	Yes	0.037
-2	Smooth	Yes	0.056

after leveling the tubes, by adjusting the leveling nuts on the test condenser support bracket. When the condensation or the inundation rate increased, it was observed that the drops were formed at more sites along the tubes, but the droplet size for each tube was nearly the same.

2. Roped Tubes

Aqain, during the experiments, there was no evidence of drop-wise condensation. It was observed that the falling drops were coalesced on the lower surface of the tube, especially in the space between the two successive grooves. and then rivulets were formed and fell on the tube below. Under condensate inundation, the phenomenon of the rivulet formation was more intense. It is also worth noting that the droplet-formation frequency was higher near the cooling water inlet end. This can be easily explained by the larger, local, Sieder-Tate coefficient and by the larger temperature difference, $T_{sat} - T_{ci}$, at the inlet compared to $T_{sat} - T_{co}$, at the outlet end.

3. Roped Tubes Wrapped with Wire

The condensate was formed in the space between the successive grooves wrapped with wire. Then due to the surface tension forces. the condensate was drawn to the base of the wire, and drops were formed at the bottom surface of the tubes.

Under inundation conditions, the above-mentioned drop formation and movement were more intense. It is worth noting that no splashing was observed during the experiment.

4. Smooth Tubes Wrapped with Wire

It was observed that the condensate forms between two successive, helically-wrapped wires, and was then drawn immediately towards the space between the wire and the tube surface. Rivulets were drawn from the base of the wire to the next tube below. It was observed that there was no drop migration along the tube.

Under condensate inundation. the above mentioned drop formation was more intensive. but again no splashing was observed.

VI. CONCLUSIONS

1. The average. outside. heat-transfer coefficient for 30 smooth tubes was 59 percent of the Nusselt coefficient calculated for the first tube in the bank.

2. The average, outside, heat-transfer coefficient for 30 smooth tubes wrapped with wire was 86 percent of the Nusselt coefficient calculated for the first tube in the bank.

3. The average. outside. heat-transfer coefficient for 30 roped tubes was 63 percent of the Nusselt coefficient calculated for the first tube in the bank.

4. The average. outside. heat-transfer coefficient for 30 roped tubes, wrapped with wire was 86 percent of the Nusselt coefficient calculated for the first tube in the bank.

5. The Sieder-Tate coefficient for roped tubes was 2.1 times greater than that for smooth tubes.

6. Wire wrapping considerably improves the average condensing coefficient for both smooth and roped tubes in tube bundles.

7. Of all cases investigated in this study, roped tubes with wire wrap would be the best candidate for designing compact condensers. However, the water side pumping power, which was not investigated in this work, could be considerably higher than that for the smooth-tube case.

VII. RECOMMENDATIONS

A. TEST APPARATUS MODIFICATIONS

The following test apparatus modifications are considered advisable:

1. Redesign the test condenser flanges to allow a large sealing area, and also to allow for the possibility of further experiments with different S/D ratios.

2. Redesign the existing test condenser hotwell to allow more reliable and convenient measurement of condensate flow rate.

3. Install a larger heater for the perforated-tube water supply tank to facilitate the simulation of larger tube bundles. Also, modify the temperature control system for the perforated-tube water supply tank, to allow for more rapid heating and cooling to reduce long delays between runs.

4. Install a glass window on the rear side of the test condenser, in order for the operator to have a view on both sides of the test tubes.

5. Make the system vacuum tight so that data can be taken at vacuum conditions.

B. ADDITIONAL TESTS

The following additional tests would be important in this continued investigation.

1. Conduct tests with enhanced tubes manufactured by Yorkshire Imperial Metals, Ltd.

2. Take movies of the condensation process so that further conclusions could be drawn with regard to condensate drop phenomena and their relationship to condenser performance.

3. Conduct tests varying the pitch of the wrapped wire on smooth titanium tubes and determine the optimum wire pitch.

4. Withdraw gas samples from the test condenser and analyze on the gas chromatograph to determine the effect of noncondensable gases on the condenser performance.

5. Conduct tests to investigate the effect of vapor velocity on the heat-transfer coefficient.

APPENDIX A

OPERATING PROCEDURES

A. INITIAL PROCEDURES

1. Energize the main circuit breaker located in Power Panel P-2 on the wall to the right of the test apparatus.

2. Energize the circuit breaker on the left side of the old control board by pressing the ON button.

3. Energize the following switches in the control panel:

   a. #1 - Perforated tube water supply tank (feed pump).

   b. #2 - Outlets.

   c. #3 - Hot water heater.

   d. #4 - Condensate pump.

   e. #6 - Cooling tower.

   f. #7 - Cooling water pump.

4. Ensure all test apparatus valves are closed.

5. Fill the perforated tube condensate supply tank with distilled water. Set the temperature controller for the perforated tube water supply tank at about 95° C, fully open the recirculation valve, P-1, and start the pump to begin heating the water. The controller will have to be reset to the proper supply temperature once steady-state conditions are obtained.

6. Fill the cooling water supply tank. This can be done by backfilling with valves CK-1 and CW-4 open.

7. Energize the data acquisition system.

B. OPERATION

1. House Steam

   a. Open the main supply valve.

   b. Open valve MS-3 until the pressure gage indicates the desired steam supply pressure.

   c. Fully open MSD-1 to drain any condensate.

   d. Open valves MS-4 and MS-5 until the desired steam supply pressure is obtained and re-adjust MS-3 as necessary.

2. Condensate System. To collect the condensate in the test condenser hotwell, operate the system with valve C-1 closed. After a test run is completed, open valve C-1 to drain the condensate by opening valve C-2 to the bilges. or operating the condensate pump in order to fill the perforated tube supply tank.

3. Cooling-Water System

   a. Open valves CW-1, CW-2 and CW-3.

   b. Ensure valves CK1-1 and CW-4 are closed.

   c. Energize the two cooling water pumps.

   d. Open valves CW-5, CW-6, CW-7, CW-8 and CW-9 to obtain the desired cooling rates.

4. Perforated Tube Water Supply System

   a. Once steady-state conditions are achieved, reset the temperature controller to the proper inundation temperature.

   b. Adjust the rotameter to the required flow rate for each run. The supply tank recirculation valve may have to be adjusted to achieve the desired flow rate, but should never be fully closed to avoid damaging the pump and to ensure uniform water temperature in the tank.

   c. Refill the supply tank as required. by using the existing piping filling system, by operating the condensate pump; or filling the supply tank with distilled water filling line.

5. Miscellaneous.

To maintain a clear test condenser window, open valve A-2 and then energize and adjust the air heater power supply. When securing, always turn off the power supply first and allow the air heater to cool before securing valve A-2.

C. SECURING THE TEST APPARATUS

1. Secure the steam valves MS-5, MS-4, MS-3 and the main supply valve.

2. Secure the air compressor.

3. Secure the perforated-tube water supply system by securing the pump, temperature controller, and valves P-1 and P-4. Drain the supply tank by opening valve P-3 (if desired).

4. Secure the test condenser viewing window air heater as prescribed above.

5. Secure the data acquisition system.

6. Allow the test condenser to cool down for about 30 minutes, then secure the cooling water pumps and close valves CS-1, CW-2, CW-3, CW-5, CW-6, CW-7, CW-8, CW-9 and C-4.

7. Drain the test condenser hotwell.

8. Secure all circuit breakers.

9. Drain the cooling water system piping and rotameters by opening valve CW-10 and leaving open the cooling water rotameter supply valves.

10. Drain the cooling water supply tank by opening the drain valve via the remote operating rod.

11. Ensure all valves are secured.

APPENDIX B

SAMPLE CALCULATIONS

A. RUN STNWNI-1

A sample calculation is performed in this section to illustrate the solution procedure used in the data reduction program [Ref 29].

The STNWNI-1 run was selected to perform this analysis:

INPUT PARAMETERS

File				STNWNI-1	
Pressure condition				Atmospheric	
Inundation condition				5 tubes	
Month, date and time				05:11:10:08:50	
Run number				1	
Tube number	1	2	3	4	5
Inlet temp (^{o}C)	28.53	28.54	28.55	28.54	28.53
Outlet temp (^{o}C)	32.55	32.20	32.17	32.05	31.83
Saturation temperature				100.24 (^{o}C)	
Degree of superheat				1.88 (^{o}C)	
Condensate temperature				92.41 (^{o}C)	
Static pressure				767.33 (mm Hg)	

The following calculations are further limited only to the first tube.

1. Determination of Average Bulk Temperature

$$T_b(1) = T_{ci}(1) + T_{co}(1) \quad x0.5$$

$$T_b(1) = (28.53 + 32.55)\ 0.5$$

$$T_b(1) = 30.54^{o}C$$

2. <u>Thermophysical Properties</u>

$$P_r = 5.329$$

$$\rho = 995.2 \text{ kg/m}^3$$

$$\mu = 798 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

$$C_{pw} = 4.178 \text{ Kj/kg.K}$$

$$k_w = 619 \times 10^{-3} \text{ W/m}\cdot\text{K}$$

NOTE: All properties are calculated at the average bulk water temperature from Table A.6 p. 782 [Ref 30]

3. <u>Cooling Water Mass Flow Rate</u>

$$\dot{m} = 14.22 \text{ Kg/min} = 0.237 \text{ Kg/s}$$

4. <u>Determination of Cooling Water Velocity</u>

$$V_w = \frac{M_f}{\rho A_i}$$

$$V_w = \frac{(14.22)\ \frac{1}{60}}{(995.2)\ (1.56 \times 10^{-4})}$$

$$V_w = 1.53 \text{ m/s}$$

5. <u>Determination of Reynolds Number</u>

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

$$R_e = \frac{(995.2)\ (1.53)\ (0.141)}{798 \times 10^{-6}}$$

$R_e = 26{,}904$

6. <u>Determination of Heat Transfer</u>

$$Q = \dot{m} \cdot (T_{co} - T_{ci}) \cdot C_{pw}$$

$$Q = (14.22)\ (\frac{1}{60})\ (32.55 - 28.53)\ (4{,}178)$$

$$Q = 3{,}980.5\ W$$

7. <u>Determination of Heat Flux</u>

$$q'' = \frac{Q}{\pi \cdot D_o \cdot L}$$

$$q'' = \frac{3{,}980.5}{\pi \cdot (0.015875)\ (0.305)}$$

$$q'' = 261{,}682.2\ \frac{W}{m^2}$$

8. <u>Determination of Nusselt Coefficient</u>

$$h_{Nu} = 0.651 \left[\frac{k_f^3 \cdot \rho_f^2 \cdot h_{fg} \cdot (9.81)}{\mu_f \cdot D_o \cdot q''}\right]^{1/3}$$

Assume $T_f = T_{sat}$

$$T_f = 100.24^{\circ}C$$

$$\rho_f = 957.8\ kg/m^3$$

$$h_{fg} = 2.2564 \times 10^3\ J/kg$$

$$k_f = 682 \times 10^{-3}\ W/m \cdot K$$

$$u_f = 281 \times 10^{-6} \frac{N_s}{m^2}$$

NOTE: All properties are calculated at the Saturation Temperature. From Table A.6 p. 782 [Ref 30]

$$h_{Nu} = 0.651 \left[\frac{(682x10^{-3})^3 \cdot (957.8)^2 (2256.4x10^3)(9.81)}{(281x10^{-6})(0.015975)(261682.2} \right]^{1/3}$$

$$h_{Nu} = 11492.6 \text{ W/m}^2\text{K}$$

9. Determination of $T_{f,c}$

$$T_{f,c} = T_{sat} - \frac{q''}{h_{Nu}} 0.5$$

$$T_{f,c} = 100.24 - \frac{261685.3}{11492.6} 0.5$$

$$T_{f,c} = 88.85^oC$$

10. Thermophysical Properties

$$k = 673 \times 10^{-3} \text{ W/mK}$$

$$\rho = 963 \text{ Kg/m}^3$$

$$\mu = 326 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

$$h_{fg} = 2.289.5 \times 10^3 \text{ J/kg}$$

NOTE: All properties are calculated at the film temperature. From Table A.6 p. 782 [Ref 30]

11. Determination of Nusselt Coefficient

$$h_{Nu} = 0.651 \left[\frac{k_f^{3} \cdot \rho_f^{2} h_{fg} \cdot 9.81}{\mu_f \cdot D_o \cdot q''}\right]^{1/3}$$

$$h_{Nu} = 0.651 \left[\frac{(673 \times 10^{-3})^{3} \cdot (963)^{2} (2289.5 \times 10^{3})(9.81)}{(326 \times 10^{-6})(0.015875)(261685.4)}\right]^{1/3}$$

$$h_{Nu} = 10{,}885.3 \ W/m^{2} \cdot K$$

12. Determination of $T_{f,c}$

$$T_{f,c} = T_{sat} - \frac{q''}{h_{Nu}} \, 0.5$$

$$T_{f,c} = 100.24 - \frac{261685.4}{10885.3} \, 0.5$$

$$T_{f,c} = 88.22^{o}C$$

13. Determination of Logarithmic Meal Temperature Difference (LMTD)

$$LMTD = \frac{T_{co} - T_{ci}}{\ell n \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}}\right)}$$

$$LMTD = \frac{32.55 - 28.53}{\ell n \left(\frac{100.24 - 28.53}{100.24 - 32.55}\right)}$$

$$LMTD = 69.68^{o}C$$

14. Determination of Overall Heat - Transfer Coefficient

$$U_o = \frac{q''}{LMTD}$$

$$U_o = \frac{261685.4}{69.68}$$

$$U_o = 3755.5 \frac{W}{m^2K}$$

15. Determination of Inside Heat-Transfer Coefficient

Assume $C_f = 1.1$

$C_i = 0.029$

$$h_i = \frac{K_w}{D_i} \cdot C_i \cdot R_e{}^{0.8} \cdot P_r{}^{0.333} \cdot C_f$$

$$h_i = \frac{0.673}{0.0141} (0.029)(26904^{0.8})(5.829^{0.3333}) \cdot 1.1$$

$$h_i = 9303.2 \ W/m^2K$$

16. Determination of Inner Wall Temperature

$$T_w = T_b + \frac{q''}{h_i} \frac{D_o}{D_i}$$

$$T_w = 30.54 + \frac{(261685.4)}{(9303.2)} \frac{(0.015875)}{(0.0141)}$$

$$T_w = 62.2^oC$$

17. Determination of μ_w at the Average Wall Temperature

$$\mu_w \ (62.2^oC) = 453 \times 10^{-6} \ N \cdot s/m^2$$

18. Determination of Correction Factor

$$C_{fc} = \left[\frac{\mu_w}{\mu_w \ (65.59^\circ)}\right]^{0.14}$$

$$C_{fc} = \left[\frac{798 \times 10^{-6}}{453 \times 10^{-6}}\right]^{0.14}$$

$$C_{fc} = 1.08$$

19. Determination of Outside Heat-Transfer Coefficient

$$h = \frac{1}{\frac{1}{U_o} - \frac{D_o}{D_i h_i} - R_w}$$

$$h = \frac{1}{\frac{1}{3755.5} - \frac{0.015875}{(0.1014)(9303.2)} - 0.000042925}$$

$$h = 9772.3 \ W/m^2K$$

B. RUN STSD-11

A sample calculation is performed in this section to illustrate the solution procedure used in the modified Wilson plot program [Ref 29].

The DP-11 run was selected to perform this analysis

INPUT PARAMETERS

File	DP-11
Month, date and time	04:01:13:57:20
Data point	#1
Steam Saturation Temperature	100.01°C
Inlet Temperature	24.42°C
Outlet Temperature	30.75°C
Flowmeter Reading	10%

1. Determiantion of Average Bulk Water Temperature

$$T_b = (T_{ci} + T_{co})\ 0.5$$

$$T_b = (24.42 + 30.75)\ 0.5 = 27.59°C$$

2. Thermophysical Properties

$$P_r = 5.83$$

$$\rho = 997\ \frac{Kg}{m^3}$$

$C_{pw} = 4.180\ Kj/Kg \cdot K$

$\mu = 857 \times 10^{-6}\ N \cdot s/m^2$

$k_w = 0.613\ W/m \cdot K$

NOTE: All properties are calculated at the average bulk water temperature from Table A.6 p. 782 [Ref. 30]

3. Cooling Water Mass Flow Rate

$M_f = 6.36\ kg/min$

4. Determination of Cooling Water Velocity

$$V_w = \frac{M_f}{\rho A_i}$$

$$V_w = \frac{(6.36)\ \frac{1}{60}}{(997)\ (1.56 \times 10^{-4}}$$

$V_w = 0.68\ m/s$

5. Determination of Reynolds Number

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

$$R_e = \frac{(997)\ (0.68)\ (0.0141)}{857 \times 10^{-6}}$$

$R_e = 11{,}154$

6. Determination of Heat Transfer

$$Q = M_f \cdot (T_{co} - T_{ci}) \cdot C_{pw}$$

$$Q = (6.36)\ (\frac{1}{G_o})\ (30.75 - 24.42)\ (4180)$$

$Q = 2{,}805\ W$

7. Determination of Logarithmic Mean Temperature Difference (LMTD)

$$LMTD = \frac{T_{co} - T_{ci}}{\ell n\left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}}\right)}$$

$$LMTD = \frac{30.75 - 24.42}{\ell n\left(\frac{100.01 - 24.42}{100.01 - 30.75}\right)}$$

$$LMTD = 72.38^{o}C$$

8. Determination of Overall Heat - Transfer Coefficient

$$U_o = \frac{Q}{\pi \cdot D_o \cdot L \cdot LMTD}$$

$$U_o = \frac{2805}{\pi(0.015875)(0.305)(72.38)}$$

$$U_o = 2{,}547.715 \ \frac{W}{m^2 \cdot K}$$

9. Determination of Sieder - Tate Parameter

$$X = R_e^{-0.8} \cdot P_r^{-0.3333}$$

$$X = (11{,}154^{-0.8}) \ (5.83^{-0.3333})$$

$$X = 0.0003212$$

10. Determination of Inside Heat - Transfer Coefficient, based on assumed Sieder - Tate Coefficient

Assume $C_f = 1.1$ and

$C_i = 0.03$

$$h_i = \frac{K_w}{D_i} C_i \cdot R_e^{0.8} \cdot P_r^{0.333} C_f$$

$$h_i = \frac{613 \times 10^{-3}}{0.141} (0.03)(11{,}154^{0.8})(5.83^{0.333})(1.1)$$

$$h_i = 4{,}463.4 \ \frac{W}{m^2 \cdot K}$$

11. Determination of Average Wall Temperature

$$T_w - T_b = \frac{Q}{\pi \cdot D_i \cdot L \cdot h_i}$$

$$T_w - T_b = \frac{2805}{\pi(0.0141)(0.305)(4{,}453.4)}$$

$$T_w - T_b = 46.51^{\circ}C$$

$$T_w = 46.51 + T_b$$

$$T_w = 46.51 + 27.59 = 74^{\circ}C$$

12. Obtain μ_w at the Average Inner Wall Temperature

$$\mu_w \ (T_w) = 375 \times 10^{-6} \ N \cdot s/m^2$$

13. Determination of C_{fc}

$$C_{fc} = \left(\frac{\mu_w}{\mu_w(T_w)}\right)^{0.14}$$

$$C_{fc} = \left(\frac{857 \times 10^{-6}}{375 \times 10^{-6}}\right)^{0.14}$$

$$C_{fc} = 1.12267$$

14. Iterate for h_i

Assume $C_f = 1.12$ and

$C_i = 0.03$

$$h_i = \frac{K_w}{D_i} C_i R_e^{0.8} P_r^{0.3333} \cdot C_f$$

$$h_i = \frac{613 \times 10^{-3}}{0.0141} \cdot (0.03)(11{,}154^{0.8})(5.83^{0.3333})(1.12)$$

$$h_i = 4{,}546.97 \text{ W/m}^2\text{K}$$

15. <u>Determination of Wall Thermal Resistance, Based on the Outside Diameter</u>

$$R_w = \frac{(D_o - D_i)\ D_o}{k_m \cdot (D_o + D_i)}$$

$$R_w = \frac{(0.015875 - 0.0141)(0.015875)}{(21.9)(0.015875 + 0.0141)}$$

$$R_w = 0.000042925 \ \frac{m^2 \cdot K}{W}$$

Assume $R_{f,} = 0$

16. <u>Determination of Outside Heat - Transfer Coefficient</u>

$$h = \frac{1}{\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}}$$

$$h = \frac{1}{\frac{1}{2{,}547.715} - 0.000042925 - \frac{0.015875}{(0.0141)(4{,}463.4)}}$$

$$h = 10273.8 \ \frac{W}{m^2 \cdot K}$$

Set $Q_o = Q = 2805$ W

17. <u>Determination of Actual Sieder-Tate Parameter</u>

$$X = \frac{X}{C_{fc}}$$

$$X = \frac{0.0003212}{1.12267}$$

$X = 0.0002861$

INPUT PARAMETERS

File	STSD-11
Data point	#2
Steam saturation temperature	100.03°C
Inlet temperature	24.39°C
Outlet temperature	29.46°C
Rotameter setting	15%

18. Determination of Average Bulk Water Temperature

$$T_b = (T_{ci} + T_{co}) \times 0.5$$

$$T_b = (24.39 + 29.46)\ 0.5$$

$$T_b = 26.92°C$$

19. Thermophysical Properties

$$P_r = 5{,}823$$

$$\mu = 996\ Kg/m^3$$

$$C_{pw} = 4.180\ Kj/KgK$$

$$\mu = 855 \times 10^{-6}\ N{\cdot}S/m^2$$

$$k_w = 613 \times 10^{-3}\ W/mK$$

20. Cooling Water Mass Flow Rate

$$M_f = 9.72\ Kg/min$$

22. Determination of Cooling Water Velocity

$$V_w = \frac{M_f}{\rho A_i}$$

$$V_w = \frac{(9.72)\ \frac{1}{60}}{(996)(1.56 \times 10^{-4})}$$

$$V_w = 1.04 \text{ m/s}$$

22. Determination of R_e

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

$$R_e = \frac{(996)(1.04)(0.0141)}{855 \times 10^{-6}}$$

$$R_e = 17{,}082$$

23. Determination of Heat - Transfer

$$Q = M_f\ (T_{co} - T_{ci})\ C_{pw}$$

$$Q = (9.72)(\frac{1}{60})(29.46 - 23.39)(4180)$$

$$Q = 3{,}433.2 \text{ W}$$

24. Determination of Logarithmic Mean Difference Temperature (LMTD)

$$\text{LMTD} = \frac{T_{co} - T_{ci}}{\ln\left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}}\right)}$$

$$\text{LMTD} = \frac{29.46 - 24.39}{\ln\left(\frac{100.03 - 24.39}{100.03 - 29.46}\right)}$$

$$\text{LMTD} = 73.08^{o}\text{C}$$

25. Determination of Overall Heat-Transfer Coefficient

$$U_o = \frac{Q}{\pi \cdot D_o \cdot L \cdot \text{LMTD}}$$

$$U_o = \frac{3433.2}{\cdot(0.015875)(0.305)(73.08)}$$

$$U_o = 3{,}088.4 \ \mathrm{W/m^2K}$$

26. <u>Determination of Sieder-Tate Parameter</u>

$$X = R_e^{-0.8} \cdot P_r^{-0.3333}$$

$$X = (17{,}082^{-0.8})\ (5.823^{-0.3333})$$

$$X = 0.000228$$

27. <u>Determination of $\frac{1}{h_i}$</u>

$$\frac{1}{h_i} = \frac{1}{U_o} - \frac{1}{h_o}\left(\frac{Q_o}{Q}\right)^{1/3} - R_w\left(\frac{D_i}{D_o}\right)$$

$$\frac{1}{h_i} = \left[\frac{1}{3088.4} - \frac{1}{10273}\left(\frac{2805}{3{,}433.2}\right)^{1/3} - 0.00004295\right]\frac{0.015875}{0.0141}$$

$$\frac{1}{h_i} = 0.0002095\ \frac{\mathrm{m^2K}}{\mathrm{W}}$$

28. <u>Determination of Average Wall Temperature</u>

$$T_w = T_b + \frac{Q}{\pi \cdot D_i \cdot L \cdot h_i}$$

$$T_w = 26.92 + \frac{3433.2}{\pi \cdot (0.015875)(0.305)(4{,}773.26)}$$

$$T_w = 74.20^{\circ}\mathrm{C}$$

29. <u>Determination of μ_w at the Average Wall Temperature</u>

$$\mu_w(T_w) = 375.2 \times 10^{-6}\ \mathrm{N \cdot s/m^2}$$

30. Determination of C_{fc}

$$C_{fc} = \left(\frac{u_w}{u_w(T_w)}\right)^{0.14}$$

$$C_{fc} = \left(\frac{855 \times 10^{-6}}{375.2 \times 10^{-6}}\right)^{0.14}$$

$$C_{fc} = 1.1222$$

31. Determination of Actual Sieder-Tate Parameter

$$X = \frac{X}{C_{fc}}$$

$$X = \frac{0.000228}{1.12459}$$

$$X = 0.0002027$$

APPENDIX C

UNCERTAINTY ANALYSIS

The general form of Kline and McClintock [Ref. 31] "second order" equation is used to compute the probable uncertainty in the results. For some resultant, R, which is a function of primary variables X_1, X_2,..., X_n, the probable uncertainty in R, δR is given by:

$$\delta R = \left[\left(\frac{\theta R}{\theta X_1}\ \delta X_1\right)^2 + \left(\frac{\theta R}{\theta X_2}\ \delta X_2\right)^2 + \ldots + \left(\frac{\theta R}{\theta X_n}\ \delta X_n\right)^2 \right]^{1/2} \qquad (C.1)$$

where δX_1, δX_2,..., δX_n is the possible uncertainty in each of the measured variables.

A. UNCERTAINTY IN THE COOLING WATER VELOCITY

$$V_w = \frac{\dot{m}}{\rho A_i}$$

Applying equation (C.1) the following equation results:

$$\frac{\delta V_w}{V_w} = \left[\left(\frac{\delta \dot{m}}{\dot{m}}\right)^2 + \left(\frac{\delta \rho}{\rho}\right)^2 + \left(\frac{\delta A_i}{A_i}\right)^2 \right]^{1/2} \qquad (C.2)$$

The following uncertainties were assigned to the variables:

$\delta \dot{m} = \pm 0.01$ kg/s

$\delta \rho = \pm 3$ kg/m^3

$\delta A_i = \pm 0.0001$ m^2

B. UNCERTAINTY IN THE REYNOLD'S NUMBER

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

The probable uncertainty is given by:

$$\frac{\delta R_e}{R_e} = \left[\left(\frac{\delta \rho}{\rho}\right)^2 + \left(\frac{\delta V_w}{V_w}\right)^2 + \left(\frac{\delta D_i}{D_i}\right)^2 + \left(\frac{\delta \mu}{\mu}\right)^2 \right]^{1/2} \qquad (C.3)$$

The following uncertainties were assigned to the variables:

$\delta\rho = \pm 3 \text{ kg/m}^3$

$\delta D_i = \pm 0.0001 \text{ m}$

$\delta\mu = \pm 8\text{x}10^{-6} \text{ N}\cdot\text{s/m}^2$

δV_w = from equation (C.2)

C. UNCERTAINTY IN HEAT TRANSFER

$$Q = \dot{m}(T_{co} - T_{ci})C_{pw}$$

The probable uncertainty is given by:

$$\frac{\delta Q}{Q} = \left[\left(\frac{\delta \dot{m}}{\dot{m}}\right)^2 + \left(\frac{\delta T_{co}}{T_{co} - T_{ci}}\right)^2 + \left(\frac{\delta T_{ci}}{T_{co} - T_{ci}}\right)^2 + \left(\frac{\delta C_{pw}}{C_{pw}}\right)^2 \right]^{1/2} \qquad (C.4)$$

The following uncertainties were assigned to the variables:

$\delta\dot{m} = \pm 0.01 \text{ kg/s}$

$\delta T_{co} = \pm 0.025 \text{ °C}$

$\delta T_{ci} = \pm 0.025 \text{ °C}$

$\delta C_{pw} = \pm 8 \text{ J/Kg}\cdot\text{C}$

D. UNCERTAINTY OF THE HEAT-FLUX

$$q'' = \frac{Q}{\pi D_o L}$$

The probable uncertainty is given by:

$$\frac{\delta q''}{q''} = \frac{1}{\pi}\left[\left(\frac{\delta Q}{Q}\right)^2 + \left(\frac{\delta D_o}{D_o}\right)^2 + \left(\frac{\delta L}{L}\right)^2\right]^{1/3} \quad (C.5)$$

The following uncertainties were assigned to the variables:

$\delta D_o = \pm 0.0001$ m

$\delta L = \pm 0.0001$ m

δQ = as found from equation (C.4)

E. UNCERTAINTY OF h_{Nu}

$$h_{Nu} = 0.651\left[\frac{k_f^3 \rho^2 h_{fg} \cdot g}{\mu_f D_o q}\right]^{1/3}$$

The probable uncertainty is given by:

$$\frac{h_{Nu}}{h_{Nu}} = 0.651\left[\left(\frac{k_f}{k_f}\right)^2 + \left(\frac{2}{3}\frac{\delta\rho}{\rho}\right)^2 + \left(\frac{\delta h_{fg}}{3h_{fg}}\right)^2 + \left(\frac{\delta g}{3g}\right)^2 + \left(\frac{\delta \mu_f}{3\mu_f}\right)^2 + \right.$$
$$\left. + \left(\frac{\delta D_o}{3D_o}\right)^2 + \left(\frac{\delta q''}{3q''}\right)^2\right]^{1/2} \quad (C.6)$$

The following uncertainties were assigned to the variables:

$\delta k_f = \pm 0.0012$ w/m·k

$\delta\rho = \pm 3$ kg/m^2

$\delta h_{fg} = \pm 0.48$ J/kg

$\delta\mu_f = \pm 8\text{x}10^{-6}$ N·S/m^2

$\delta D_o = \pm 0.0001$ m

$\delta g = \pm 0.001$ m/s^2

$\delta q''$ = as found from equation (C.5)

F. UNCERTAINTY OF T_{film_c}

$$T_{f_c} = T_{sat} - \frac{q''}{h_{Nu}} 0.5$$

The probable uncertainty is given by:

$$\frac{\delta T_{f_c}}{T_{f_c}} = \left[\left(\frac{\delta T_{sat}}{T_{sat}}\right)^2 + \left(\frac{\delta q''}{q''}\right)^2 + \left(\frac{\delta h_{Nu}}{h_{Nu}}\right)^2 \right]^{1/2} \qquad \text{(C.7)}$$

The following uncertainties were assigned to the variables:

$\delta T_{sat} = \pm 0.025$ °C

$\delta q''$ = as found from equation (C.5)

δh_{Nu} = as found from equation (C.6)

G. UNCERTAINTY OF OVERALL HEAT-TRANSFER COEFFICIENT

$$U_o = \frac{q''}{LMTD}$$

The probable uncertainty is given by:

$$\frac{\delta U_o}{U_o} = \left[\left(\frac{\delta q''}{q''}\right)^2 + \left(\frac{\delta (LMTD)}{LMTD}\right)^2 \right]^{1/2} \qquad \text{(C.8)}$$

The following uncertainties were assigned to the variables:

δq = as found from equation (C.5)

$\delta(LMTD)$ = as found from equation (C.9)

H. UNCERTAINTY FOR LOGARITHMIC MEAN TEMPERATURE DIFFERENCE (LMTD)

$$LMTD = \frac{T_{co}-T_{ci}}{\ln\left(\frac{T_s-T_{ci}}{T_s-T_{co}}\right)}$$

The probable uncertainty is given by:

$$\frac{\delta LMTD}{LMTD} = \left[\left(\frac{\delta T_s (T_{ci}-T_{co})}{(T_s-T_{ci})(T_s-T_{co}) \quad \ln\left(\frac{T_s-T_{ci}}{T_s-T_{co}}\right)}\right)^2 + \left(\frac{\delta T_{ci}}{(T_s-T_{ci}) \quad \ln\left(\frac{T_s-T_{ci}}{T_s-T_{co}}\right)}\right)^2 + \left(\frac{\delta T_{co}}{(T_s-T_{co}) \quad \ln\left(\frac{T_s-T_{ci}}{T_s-T_{co}}\right)}\right)^2\right]^{1/2} \quad (C.9)$$

The following uncertainties were assigned to the variables:

$\delta T_s = \pm 0.025$ °C

$\delta T_{ci} = \pm 0.025$ °C

$\delta T_{co} = \pm 0.025$ °C

I. UNCERTAINTY OF INSIDE HEAT-TRANSFER COEFFICIENT

$$h_i = \frac{kw}{D_i} \cdot C_i R_e^{0.8} P_r^{0.333} \left(\frac{\mu}{\mu w}\right)^{0.14}$$

The probable uncertainty is given by:

$$\frac{\delta h_i}{h_i} = \left[\left(\frac{\delta k_w}{k_w}\right)^2 + \left(\frac{\delta D_i}{D_i}\right) + \left(\frac{0.8\delta R_e}{R_e}\right)^2 + \left(\frac{0.333\delta P_r}{P_r}\right)^2 + \left(\frac{\delta C_i}{C_i}\right)^2 + \right.$$

$$\left. + \left(\frac{0.14\delta(\mu/\mu_m)}{\mu/\mu_m}\right)^2 \right]^{1/2} \qquad (C.10)$$

The following uncertainties were applied to the variables:

$\delta k_w = \pm 0.0012$ w/m·k

$\delta D_i = \pm 0.0001$ m

$\delta C_i = \pm 0.0001$

$\delta R_e =$ as found from equation (C.3)

$\delta P_r = \pm 0.17$

$\delta(\mu/\mu_w) = 8x10^{-6}$ Ns/m^2

J. UNCERTAINTY IN TEMPERATURE DIFFERENCE

$$\Delta T = \frac{q''}{h_i} \frac{D_o}{D_i}$$

$$\frac{\delta \Delta T}{\Delta T} = \left[\left(\frac{\delta q''}{q''}\right)^2 + \left(\frac{\delta h_i}{h_i}\right)^2 + \left(\frac{\delta D_o}{D_o}\right)^2 + \left(\frac{\delta D_i}{D_i}\right)^2 \right]^{1/2} \qquad (C.11)$$

The following uncertainties were assigned to the variables:

$\delta q'' =$ as found from equation (C.5)

$\delta h_i =$ as found from equation (C.10)

$\delta D_o = \pm 0.0001$ m

$\delta D_i = \pm 0.0001$ m

K. UNCERTAINTY IN OUTSIDE HEAT-TRANSFER COEFFICIENT

$$h_o = \frac{1}{\frac{1}{U_o} - \frac{D_o}{D_i h_i} - R_w}$$

$$\frac{\delta h_o}{h_o} = \left[\left(\frac{\delta U_o}{U_o^2 \left(\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}\right)}\right)^2 + \left(\frac{\delta R_w}{\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}}\right)^2 + \left(\frac{\left(\frac{D_o}{D_i h_i}\right)\left(\frac{\delta h_i}{h_i}\right)}{\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}}\right)^2\right]^{1/2} \qquad (C.12)$$

The following uncertainties were assigned to the variables:

δU_o = as found from equation (C.8)

δh_i = as found from equation (C.10)

$\delta R_w = \pm 0.00001 \ m^2 \cdot k/w$

L. UNCERTAINTIES FOR THE NORMALIZED LOCAL HEAT-TRANSFER COEFFICIENT h_N/h_i

This ratio is simply the heat transfer coefficient of a given tube, N, divided by that of the first tube, i.e., for the fifth tube, N=5 and:

$$\frac{h_N}{h_1} = \frac{h_5}{h_1}$$

An application of equation (C.1) results in the following equation:

$$\frac{\delta(h_N/h_i)}{(h_N/h_i)} = \left[(\frac{\delta h_i}{h_i})^2 + (\frac{\delta h_N}{h_N})^2\right]^{1/2} \tag{C.13}$$

Note: Equation (C.13) is valid only for N7/2. For example:

$$\frac{\delta(h_2/h_1)}{h_2/h_1} = \left[(\frac{\delta h_1}{h_i})^2 + (\frac{\delta h_2}{h_2})^2\right]^{1/2}$$

M. UNCERTAINTY IN THE NORMALIZED AVERAGE HEAT-TRANSFER COEFFICIENT, $\bar{h}_N/h_1$

The normalized average heat-transfer coefficient is obtained for the Nth tube by taking the average of the heat-transfer coefficients of the first N tubes and dividing this by the heat-transfer coefficient of the first tube:

$$\frac{\bar{h}_N}{h_1} = \frac{(h_1 + h_2 + \ldots + h_N)N}{h_1}$$

Applying equation (C.1) to the above, the following equation results:

$$\frac{(\bar{h}_N/h_1)}{(\bar{h}_N/h_1)} = \left[\sum_{i=1}^{N} \frac{\delta h_i}{\sum_{i=1}^{N} h_i}^{2} + (\frac{\delta h_1}{h_1})^2\right]^{1/2} \tag{C.14}$$

where N = the tube number. For example:

$$\frac{(\bar{h}_2/h_1)}{(\bar{h}_2/h_1)} = \left[(\frac{\delta h_1}{h_1+h_2})^2 + (\frac{\delta h_2}{h_1+h_2})^2 + (\frac{\delta h_1}{h_1})^2\right]^{1/2}$$

For Run STNWNI-1

$V_w = 1.54 \pm 0.11\ ^m/s$

$Re = 27546 \pm 2121$

$Q = 3934 \pm 169\ W$

$q'' = 267830 \pm 11516\ ^W/_{m^2}$

$h_{Nu} = 10802 \pm 164\ ^W/_{m^2 \cdot k}$

$LMTD = 69.7 \pm 0.6\ °C$

$U_o = 3843.7 \pm 165\ ^W/_{m^2 \cdot k}$

$h_i = 9381 \pm 122\ ^W/_{m^2 \cdot k}$

$h_o = 11361.4 \pm 1249\ ^W/_{m^2 \cdot k}$

$^{h_2}/_{h_1} = 0.8891 \pm 0.14$

$^{\bar{h}_2}/_{h_1} = 0.9446 \pm 0.12$

APPENDIX D

COMPUTER PROGRAMS

A. DATA REDUCTION PROGRAM

```
1000! FILE NAME: DRP
1010! REVISED:   May 20, 1983
1020  COM /Ci/ C(7)
1030  DIM Tci(2),Tco(4,2),Ti(4),Mft(4),Vw(4),Ho(4)
1040  DIM To(4),Ts(1),Tb(4),R3(4),R4(4),S3(4),S4(4)
1050!
1060! ASSIGN COEFFICIENTS FOR THE 8-TH ORDER
1070! POLYNOMIAL FOR TYPE-T (COPPER-CONSTANTAN)
1080! THERMOCOUPLES
1090  DATA 0.10086091,25727.94369,-767345.8295,78025595.81
1100  DATA -9247486589,6.97688E+11,-2.66192E+13,3.94078E+14
1110  READ C(*)
1120!
1130! ASSIGN FULL-SCALE FLOW RATES THROUGH THE 5
1140! FLOW METERS (kg/min)
1150  DATA 56.86,73.35,72.44,72.52,72.24
1160  READ Mft(*)
1170!
1180! ASSIGN SIEDER-TATE COEFFICIENT AND EXPONENT
1190! FOR REYNOLDS NUMBER
1200  Ci=.029
1210  Ex=.8
1220!
1230! ASSIGN GEOMETRIC VARIABLES
1240  Di=.0141        ! Inner diameter (m)
1250  Do=.015875      ! Outer diameter (m)
1260  Ktm=21.9        ! Thermal conductivity of titanium (W/m-K)
1270  L=.305          ! Condensing length (m)
1280  Nc=3.3333       ! Number of unit cells across condenser width
1290  Pt=1.5          ! Transverse tube pitch-to-diameter ratio
1300!
1310! COMPUTE THE MINIMUM STEAM FLOW AREA IN THE TEST CONDENSER (m^2)
1320  Amf=Nc*Do*(Pt-PI/(4*Pt))*L
1330!
1340! COMPUTE INSIDE AREA AND WALL RESISTANCE
1350  Ai=PI*Di^2/4
1360  Rw=Do*LOG(Do/Di)/(2*Ktm)
1370!
1380  PRINTER IS 701
1390  CLEAR 709
1400  BEEP
1410  INPUT "ENTER MONTH, DATE, AND TIME (MM:DD:HH:MM:SS)",Time$
1420  OUTPUT 709;"TD";Time$
1430  BEEP
1440  INPUT "ENTER THE INPUT MODE (1=3054A,2=FILE)",Im
1450  IF Im=2 THEN
1460  BEEP
1470  INPUT "ENTER THE NAME OF THE EXISTING DATA FILE",Olddata$
1480  PRINT USING "10X,""This analysis was performed for data stored in file "",
10A";Olddata$
1490  ASSIGN @File2 TO Olddata$
1500  END IF
1510  IF Im=1 THEN
1520  BEEP
1530  INPUT "GIVE A NAME FOR THE DATA FILE TO BE CREATED",Newdata$
1540  CREATE BDAT Newdata$,20
1550  ASSIGN @File1 TO Newdata$
```

```
1560  END IF
1570  BEEP
1580  INPUT "GIVE A NAME FOR THE OUTPUT FILE",File_out$
1590  BEEP
1600  INPUT "ENTER THE PRESSURE CONDITION (1=ATM.2=VACUUM)",Mp
1610  IF Mp=1 THEN PRINT "          Pressure condition: ATMOSPHERIC"
1620  IF Mp=0 THEN PRINT "          Pressure condition: VACUUM"
1630  BEEP
1640  INPUT "ENTER THE INUNDATION CONDITION (1=5 TUBES. 2=30 TUBES)",Mi
1650  IF Mi=2 THEN PRINT "          Inundation condition: 30 TUBES"
1660  IF Mi=1 THEN PRINT "          Inundation condition:  5 TUBES"
1670  CREATE BDAT File_out$,6
1680  ASSIGN @File3 TO File_out$
1690  Ja=0
1700  Nrun=0
1710  FOR I=0 TO 4
1720  S3(I)=0.
1730  S4(I)=0.
1740  NEXT I
1750 Repeat: !
1760  Nrun=Nrun+1
1770  OUTPUT 709:"TD"
1780  ENTER 709:Time$
1790  PRINT " "
1800  PRINT USING "10X,""Month, date, and time: "",15A";Time$
1810  IF Im=2 THEN Rdf
1820  BEEP
1830  INPUT "ENTER FLOW METER READINGS (AS PERCENTAGES)",Fm1,Fm2
1840  IF Nrun MOD 5=1 AND Mi=2 AND Nrun>5 THEN
1850  BEEP
1860  INPUT "ENTER FLOW RATE FOR POROUS TUBE (AS A PERCENT)",Fpt
1870  OUTPUT @File1;Fpt
1880  Mpt=-8.361613+10.076742*Fpt
1890  END IF
1900  DISP "START COLLECTING CONDENSATE"
1910  BEEP
1920  WAIT 20
1930  OUTPUT 709:"AR AF0 AL19"
1940  OUTPUT 722:"F1 R1 T1 Z1 FL1"
1950!
1960! READ INLET WATER TEMPERATURES
1970!
1980  FOR I=0 TO 2
1990  OUTPUT 709:"AS SA"
2000  ENTER 722:Tci(I)
2010  CALL Tvsv(Tci(I))
2020  Tci(I)=FNTemp(Tci(I),I)
2030  NEXT I
2040!
2050! READ OUTLET WATER TEMPERATURES
2060!
2070  Ii=2
2080  FOR I=0 TO 4
2090  IF I=0 OR I=3 THEN
2100  Iu=2
2110  ELSE
2120  Iu=1
2130  END IF
2140  FOR J=0 TO Iu
2150  Ii=Ii+1
2160  OUTPUT 709:"AS SA"
```

```
2170  ENTER 722:Tco(I,J)
2180  CALL Tvsv(Tco(I,J))
2190  Tco(I,J)=FNTemp(Tco(I,J),Ii)
2200  NEXT J
2210  NEXT I
2220!
2230! READ STEAM TEMPERATURES
2240!
2250  FOR I=15 TO 16
2260  OUTPUT 709:"AS SA"
2270  ENTER 722:Ts(I-15)
2280  CALL Tvsv(Ts(I-15))
2290  Ts(I-15)=FNTemp(Ts(I-15),I)
2300  NEXT I
2310!
2320! READ CONDENSATE TEMPERATURE
2330!
2340  OUTPUT 709:"AS SA"
2350  ENTER 722:Tcon
2360  CALL Tvsv(Tcon)
2370  Tcon=FNTemp(Tcon,17)
2380!
2390! READ VAPOR TEMPERATURE
2400!
2410  OUTPUT 709:"AS SA"
2420  ENTER 722:Tv
2430  CALL Tvsv(Tv)
2440  Tv=FNTemp(Tv,18)
2450!
2460! READ VAPOR PRESSURE
2470!
2480  OUTPUT 709:"AS SA"
2490  ENTER 722:P_volts
2500!
2510! COMPUTE AVERAGE WATER TEMPERATURES AT INLET
2520!
2530  Ti(0)=Tci(0)
2540  Ti(1)=(Tci(0)+Tci(1))*.5
2550  Ti(2)=Tci(1)
2560  Ti(3)=(Tci(1)+Tci(2))*.5
2570  Ti(4)=Tci(2)
2580!
2590! COMPUTE AVERAGE WATER TEMPERATURES AT OUTLET
2600!
2610  FOR I=0 TO 4
2620  IF I=0 OR I=3 THEN
2630  To(I)=(Tco(I,0)+Tco(I,1)+Tco(I,2))*.3333
2640  ELSE
2650  To(I)=(Tco(I,0)+Tco(I,1))*.5
2660  END IF
2670  NEXT I
2680  Tsa=(Ts(0)+Ts(1))*.5
2690  Pvap=FNPvsv(P_volts)
2700  Tsat=FNTvsp(Pvap)
2710  Dsup=Tv-Tsat
2720!
2730! READ INFORMATION FOR CONDENSATE FLOW RATE
2740!
2750  BEEP
2760  INPUT "ENTER INITIAL AND FINAL LEVELS IN HOT WELL 1",H1,H2
```

```
2770  Dh=H2-H1
2780  IF Nrun MOD 5=1 THEN Msum=0
2790  Mf1=540.4836*Dh
2800  Md1=Mf1*FNRhow(Tsat-10)*1.0E-6/60
2810  Msum=Msum+Mf1
2820  IF Mi=2 AND Nrun<>30 AND Nrun MOD 5=0 THEN
2830  Mave=Msum/5
2840  Set=(Mave*FNRhow(Tsat-10)/10^6+.03238)/.042132
2850  END IF
2860!
2870 Rdf: !
2880!
2890  PRINT USING "10X,""Run number = "",DD";Nrun
2900  PRINT "              Tube #               :   1        2        3        4        5"
2910  IF Im=2 THEN
2920  IF Nrun MOD 5=1 AND Mi=2 AND Nrun>5 THEN ENTER @File2:Fot
2930  ENTER @File2:Ti(*),To(*),Tsa,Tcon,Tv,Pvap,Tsat,Dsup,Fm1,Fm2
2940  ENTER @File2:H1,H2
2950  END IF
2960  IF Im=1 THEN OUTPUT @File1:Ti(*),To(*),Tsa,Tcon,Tv,Pvap,Tsat,Dsup,Fm1,Fm2
2970  PRINT USING "10X,""Inlet temp (Deg C) :"",5(DDD.DD,2X)";Ti(*)
2980  PRINT USING "10X,""Outlet temp (Deg C):"",5(DDD.DD,2X)";To(*)
2990  PRINT USING "10X,""Saturation temperature = "",3D.DD,"" (Deg C)""""";Tsat
3000  PRINT USING "10X,""Degree of superheat    = "",3D.DD,"" (Deg C)""""";Dsup
3010  PRINT USING "10X,""Condensate temperature = "",3D.DD,"" (Deg C)""""";Tcon
3020  PRINT USING "10X,""Static pressure        = "",3D.DD,"" (mm Hg)""""";Pvap
3030  IF Im=1 THEN OUTPUT @File1:H1,H2
3040!
3050! CALCULATE AVERAGE BULK TEMPERATURES
3060!
3070  FOR I=0 TO 4
3080  Tb(I)=(Ti(I)+To(I))*.5
3090  NEXT I
3100!
3110  IF Mi=1 OR (Mi=2 AND Nrun<6) THEN As1=0.
3120  IF Mi=2 AND Nrun>5 THEN S1=As1
3130  S1=As1
3140  FOR J=0 TO 4
3150  IF J=0 THEN Cwf=Fm1
3160  IF J=1 THEN Cwf=Fm2
3170  Mf=Mft(J)*Cwf/(100*60)
3180  Tx=Tb(J)
3190  Vw(J)=Mf/(FNRhow(Tx)*Ai)
3200!
3210! CALCULATE INSIDE AND OUTSIDE COEFFICIENTS
3220!
3230  Rew=FNRhow(Tx)*Vw(J)*Di/FNMuw(Tx)
3240  Cf=1.
3250  Q=Mf*FNCpw(Tx)*(To(J)-Ti(J))
3260  Qp=Q/(PI*Do*L)
3270  IF (Mi=1 OR (Mi=2 AND Nrun<6)) AND J=0 THEN
3280  Tfilm=Tsat
3290  Kf=FNKw(Tfilm)
3300  Rhof=FNRhow(Tfilm)
3310  Hfg=FNHfg(Tsat)*1000
3320  Muf=FNMuw(Tfilm)
3330  Hnu=.651*(Kf^3*Rhof^2*Hfg*9.81/(Muf*Do*Qp))^.3333
3340  Tfilmc=Tsat-Qp/Hnu*.5
3350  IF ABS((Tfilmc-Tfilm)/Tfilmc)>.01 THEN
3360  Tfilm=Tfilmc
```

```
3370  GOTO 3290
3380  END IF
3390  PRINT USING "10X,""Nusselt coefficient for first tube = "",5D.D,"" (W/m^2)
"""";Hnu
3400  END IF
3410  IF J=0 THEN
3420  IF Mi=1 AND Nrun=1 THEN Ho1=0.
3430  PRINT "              Tube      Vw     Heat flux    Cond coef      R1       R2
 RR"
3440  PRINT "                 #      (m/S)     (W/m^2)      (W/m^2-K)"
3450  END IF
3460  Muw=FNMuw(Tx)
3470  Hi=FNKw(Tx)/Di*Ci*Rew^Ex*(FNPrw(Tx))^.3333*Cf
3480  Dt=Qp/Hi*Do/Di
3490  Cfc=(Muw/(FNMuw(Tx+Dt)))^.14
3500  IF ABS((Cf-Cfc)/Cfc)>.01 THEN
3510  Cf=(Cf+Cfc)*.5
3520  GOTO 3470
3530  END IF
3540  Lmtd=(To(J)-Ti(J))/LOG((Tsat-Ti(J))/(Tsat-To(J)))
3550  Uo=Qp/Lmtd
3560  Ho(J)=1./(1./Uo-Do/(Di*Hi)-Rw)
3570  Rr=Uo/Ho(J)
3580  S1=S1+Ho(J)
3590  IF Nrun MOD 5=1 THEN
3600  IF Mi=1 OR (Mi=2 AND Nrun=31) THEN Ja=0
3610  IF Mi=2 AND 5<Nrun AND Nrun<30 THEN Ja=Nrun-1
3620  IF Mi=2 AND 35<Nrun THEN Ja=Nrun-1
3630  END IF
3640  IF Mi=1 OR (30<Nrun AND Nrun<36 AND Mi=1) OR Nrun<6 THEN
3650  R1=Ho(J)/Ho(0)
3660  R2=S1/((J+1+Ja)*Ho(0))
3670  ELSE
3680  R1=Ho(J)/Ho1
3690  R2=S1/((J+1+Ja)*Ho1)
3700  END IF
3710!
3720! PRINT RESULTS
3730!
3740  PRINT USING "11X,DD,4X,DD.DD,2X,2(D.5DE,2X),3(Z.4D,2X)";J+1+Ja,Vw(J),Qp,Ho
(J),R1,R2,Rr
3750!
3760!
3770!
3780  IF Mi=2 AND Nrun<6 AND J=0 THEN
3790  Ho1=Ho1+Ho(0)/5
3800  END IF
3810  FOR K=0 TO 4
3820  IF K=J THEN S3(K)=S3(K)+R1
3830  IF K=J THEN S4(K)=S4(K)+R2
3840  NEXT K
3850  NEXT J
3860  IF Nrun MOD 5=0 THEN
3870  FOR K=0 TO 4
3880  R3(K)=S3(K)/5
3890  R4(K)=S4(K)/5
3900  S3(K)=0.
3910  S4(K)=0.
3920  NEXT K
3930  IF Mi=2 AND Nrun MOD 5=0 AND Nrun<>30 THEN As1=Nrun*R4(4)*Ho1
3940!
```

```
3950! PRINT AVERAGE RATIOS
3960!
3970  PRINT " "
3980  PRINT "             Tube #     R3          R4"
3990  FOR J=1 TO 5
4000  PRINT USING "12X,DD,2(4X,Z.4D)":J+Ja,R3(J-1),R4(J-1)
4010  OUTPUT @File3:J+Ja,R3(J-1),R4(J-1)
4020  NEXT J
4030  PRINT " "
4040  END IF
4050  IF Nrun MOD 5=0 AND Mi=2 AND Nrun<>30 AND Im=1 THEN
4060  BEEP
4070  PRINT USING "10X,""Set porous-tube flowmeter reading to "",3D.D,"" PERCENT
"""":Set
4080  END IF
4090!
4100!
4110!
4120  IF Im=1 THEN
4130  BEEP
4140  INPUT "DO YOU HAVE ANY MORE DATA (1=YES,0=NO)?",Go_on
4150  IF Go_on=1 THEN Repeat
4160  ELSE
4170  IF Mi=2 AND Nrun<30 THEN Repeat
4180  IF Mi=1 AND Nrun<10 THEN Repeat
4190  END IF
4200  IF Im=1 THEN PRINT USING "10X,DD,"" Data runs were stored in file "",10A":
Nrun,Newdata$
4210  ASSIGN @File1 TO *
4220  ASSIGN @File2 TO *
4230  ASSIGN @File3 TO *
4240  END
4250!
4260! THIS SUROUTINE CONVERTES THERMOCOUPLE VOLTAGE INTO TEMPERATURE
4270!
4280  SUB Tvsv(T)
4290  COM /Ci/ C(7)
4300  Sum=0.
4310  FOR I=0 TO 7
4320  Sum=Sum+C(I)*T^I
4330  NEXT I
4340  T=Sum
4350  SUBEND
4360!
4370! THIS FUNCTION CALCULATES PRANDTL NUMBER OF WATER IN THE
4380! RANGE 15 TO 45 DEG C
4390!
4400  DEF FNPrw(T)
4410  Y=10^(1.09976605-T*(1.3749326E-2-T*(3.968875E-5-3.45026E-7*T)))
4420  RETURN Y
4430  FNEND
4440!
4450! THIS FUNCTION CALCULATES THERMAL CONDUCTIVITY OF WATER
4460! IN THE RANGE OF 15 TO 105 DEG C
4470!
4480  DEF FNKw(T)
4490  Y=.5625894+T*(2.2964546E-3-T*(1.509766E-5-4.0581652E-8*T))
4500  RETURN Y
4510  FNEND
4520!
```

```
4530! THIS FUNCTION CALCULATES SPECIFIC HEAT OF WATER
4540! IN THE RANGE 15 TO 45 DEG C
4550!
4560  DEF FNCpw(T)
4570  Y=(4.21120858-T*(2.26826E-3-T*(4.42361E-5+2.71428E-7*T)))*1000
4580  RETURN Y
4590  FNEND
4600!
4610! THIS FUNCTION CALCULATES DENSITY OF WATER IN THE
4620! RANGE 15 TO 105 DEG C
4630!
4640  DEF FNRhow(T)
4650  Ro=999.52946+T*(.01269-T*(5.482513E-3-T*1.234147E-5))
4660  RETURN Ro
4670  FNEND
4680!
4690! THIS FUNCTION APPLIES CORRECTIONS TO THERMOCOUPLE READINGS
4700!
4710  DEF FNTemp(T,I)
4720  DIM A(14),B(14)
4730  DATA 0.640533,0.573054,0.593101,0.57298,0.56228,0.567384,0.569577
4740  DATA 0.553951,0.552008,0.566955,0.520998,0.522661,0.531008,0.560783,0.5524
05
4750  DATA 11.8744,8.63163,9.39412,8.570246,8.299436,8.36677,8.04507,7.459766
4760  DATA 7.498928,7.9408,5.87072,5.391556,6.13399,6.48586,6.326224
4770  READ A(*),B(*)
4780  IF I<15 THEN
4790  T=T-(A(I)-B(I)*.001*T)
4800  ELSE
4810  T=T-.5
4820  END IF
4830  RETURN T
4840  FNEND
4850!
4860! THIS FUNCTION COMPUTES THE SPECIFIC VOLUME OF STEAM
4870!
4880  DEF FNVvst(T)
4890  V=58.4525588-T*(1.51508776-T*(.01372746585-T*4.25366711E-5))
4900  RETURN V
4910  FNEND
4920!
4930! THIS FUNCTION CONVERTS THE VOLTAGE READING OF THE PRESSURE
4940! TRANSIDUCER INTO PRESSURE IN MM HG
4950!
4960  DEF FNPvsv(V)
4970  Y=1.1103462+163.36413*V
4980  RETURN Y
4990  FNEND
5000!
5010! THIS FUNCTION CALCULATES THE SATURATION TEMPERATURE OF STEAM AS A FUNCTION
5020! OF PRESSURE
5030!
5040  DEF FNTvsp(P)
5050  IF P<600 THEN
5060  T=31.8776158+P*(.235854929-P*(3.6613664E-4-P*2.41652372E-7))
5070  ELSE
5080  T=59.36562+P*(.07379467-P*(3.15662E-5-P*6.27246E-9))
5090  END IF
5100  RETURN T
5110  FNEND
```

```
5120!
5130! THIS FUNCTION COMPUTES THE VISCOSITY OF WATER
5140!
5150  DEF FNMuw(T)
5160  Mu=1.57609473E-3-T*(3.51198576E-5-T*(3.5835816E-7-1.365586115E-9*T))
5170  RETURN Mu
5180  FNEND
5190!
5200! THIS FUNCTION COMPUTES THE LATENT HEAT OF VAPORIZATION
5210!
5220  DEF FNHfg(T)
5230  Hfg=2497.7389-T*(2.2074+T*(1.7079E-3-2.8593E-6*T))
5240  RETURN Hfg
5250  FNEND
```

B. MODIFIED WILSON-PLOT METHOD

```
1000!*****************************************************
1010! FILE NAME: WILSON                                   *
1020!                                                     *
1030! THIS PROGRAM COMPUTES THE SIEDER-TATE                *
1040! COEFFICIENT FOR FLOW IN TUBES                       *
1050!*****************************************************
1060  COM /Cc/ C(7)
1070  DIM Vw(18),Tci(18),Tco(18),Ts(18),Md(18),Ta(18)
1080  DATA 0.10086091,25727.94369,-767345.8295,78025535.81
1090  DATA -9247486589,6.97688E+11,-2.66192E+13,3.94078E+14
1100  READ C(*)
1110  PRINTER IS 701
1120  BEEP
1130  CLEAR 709
1140  INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)",B$
1150  OUTPUT 709:"TD":B$
1160  Di=.0141
1170  Ai=PI*Di^2/4
1180  Do=.015875
1190  L=.305
1200  Km=21.9
1210  Rw=(Do-Di)*Do/(Km*(Do+Di))
1220  Rfi=0.
1230 Series: !
1240  OUTPUT 709:"TD"
1250  ENTER 709:A$
1260  PRINT USING "10X,""Month, date and time: "",15A":A$
1270  BEEP
1280  INPUT "ENTER INITIAL GUESS FOR SIEDER-TATE COEFFICIENT",Ci
1290  BEEP
1300  INPUT "ENTER EXPONENT FOR REYNOLDS NUMBER",Xn
1310  PRINT USING "10X,""Initial guess for Sieder-Tate coefficient = "",Z.DD":Ci
1320  PRINT USING "10X,""Exponent for the Reynolds number          = "",Z.DD":Xn
1330  BEEP
1340  INPUT "ENTER THE INPUT MODE (1=3054A,2=FILE)",Im
1350  IF Im=1 THEN
1360  BEEP
1370  INPUT "GIVE A NAME FOR THE DATA FILE",D_file$
1380  CREATE BDAT D_file$,5
1390  ELSE
1400  BEEP
1410  INPUT "GIVE THE NAME OF THE DATA FILE",D_file$
1420  PRINT USING "10X,""Following analysis was performed for data stored in fil
e "",10A":D_file$
1430  BEEP
1440  INPUT "ENTER THE NUMBER OF DATA RUNS STORED",Nrun
1450  END IF
1460  ASSIGN @File TO D_file$
1470  BEEP
1480  INPUT "GIVE A NAME FOR PLOTTING DATA FILE",Dplot$
1490  CREATE BDAT Dplot$,5
1500  ASSIGN @Filep TO Dplot$
1510  BEEP
1520  INPUT "ENTER ANALYSIS TYPE (1=HI,2=UI)",It
1530  IF It=1 THEN PRINT USING "10X,""Analysis type = Hi-METHOD"""
1540  IF It=2 THEN PRINT USING "10X,""Analysis type = Ui-METHOD"""
```

```
1550  K=0
1560  PRINT " "
1570  PRINT "          Data   Vw     Tsat      Ti       To"
1580  PRINT "           #   (m/S)   (C)       (C)      (C)"
1590 Repeat:!
1600!
1610! RECORDS THERMOCOUPLE AND PRESSURE TRANSDUCER
1620! READINGS AUTOMATICALLY THROUGH THE HP 3054A
1630! AUTOMATIC DATA ACQUISITION/CONTROL SYSTEM
1640!
1650  IF Im=1 THEN
1660  BEEP
1670  INPUT "ENTER FLOWMETER READING (AS A PERCENT)",Fm
1680  OUTPUT 709:"AR AF0 AL0"
1690  OUTPUT 722:"F1 R1 T1 Z1 FL1"
1700  OUTPUT 709:"AS SA"
1710! READS THERMOCOUPLE FOR WATER INLET
1720  ENTER 722:T(0)
1730  OUTPUT 709:"AR AF3 AL5"
1740  OUTPUT 722:"F1 R1 T1 Z1 FL1"
1750  FOR I=1 TO 3
1760  OUTPUT 709:"AS SA"
1770! READS THREE THERMOCOUPLES FOR WATER OUTLET
1780  ENTER 722:T(I)
1790  NEXT I
1800  OUTPUT 709:"AR AF19 AL19"
1810  OUTPUT 722:"F1 R1 T1 Z1 FL1"
1820  OUTPUT 709:"AS SA"
1830! READS PRESSURE TRANSDUCER
1840  ENTER 722:P_volts
1850  Pvap=FNPvsv(P_volts)
1860  Ts(K)=FNTvsp(Pvap)
1870  Sum=0.
1880  FOR I=0 TO 3
1890! CONVERT VOLTAGE READINGS TO TEMPERATURE
1900  CALL Tvsv(T(I))
1910  IF I=0 THEN
1920  Tci(K)=FNTemp(T(0),0)
1930  ELSE
1940  M=I+2
1950! APPLY THERMOCOUPLE CORRECTIONS
1960  To=FNTemp(T(I),M)
1970  Sum=Sum+To
1980  END IF
1990  NEXT I
2000  Tco(K)=Sum/3.
2010  ELSE
2020  ENTER @File:Ts(K),Tci(K),Tco(K),Fm
2030  END IF
2040  Ta(K)=(Tci(K)+Tco(K))*.5
2050  Md(K)=66.36*Fm/(100*60)
2060! COMPUTE WATER-SIDE VELOCITY
2070  Vw(K)=Md(K)/(FNRhow(Ta(K))*Ai)
2080  BEEP
2090  IF Im=1 THEN INPUT "ARE YOU TAKING MORE DATA (1=YES,0=NO)?",Go_on
2100  M=K+1
2110  PRINT USING "10X,DD,5(2X,DDD.DD)":M,Vw(K),Ts(K),Tci(K),Tco(K)
2120  IF Im=1 THEN OUTPUT @File:Ts(K),Tci(K),Tco(K),Fm
2130  K=K+1
2140  IF Im=1 THEN
```

```
2150  IF Go_on=1 THEN Repeat
2160  ELSE
2170  IF M<Nrun THEN Repeat
2180  END IF
2190  K=K-1
2200  J=0
2210  Jj=0
2220!
2230! PERFORM ITERATION TO COMPUTE SIEDER-TATE COEFFICIENT
2240!
2250  Ssq=0
2260  Sx=0.
2270  Sy=0.
2280  Sxs=0.
2290  Sxy=0.
2300  J=J+1
2310  PRINT " "
2320  PRINT "              Iteration number = ";J
2330  IF J=1 OR Jj=1 THEN PRINT "                    X          1/HI      (TW-TB)      Q
       CF"
2340  FOR I=0 TO K
2350  Q=Md(I)*(Tco(I)-Tci(I))*4180.
2360  Lmtd=(Tco(I)-Tci(I))/LOG((Ts(I)-Tci(I))/(Ts(I)-Tco(I)))
2370  Un=Q/(Lmtd*PI*Do*L)
2380  Unr=1./Un
2390  Rei=FNRhow(Ta(I))*Vw(I)*Di/FNMu(Ta(I))
2400  Prw=FNPrw(Ta(I))
2410  X=Rei^(-Xn)*Prw^(-.3333)
2420  IF It=2 THEN
2430  Hir=Unr*Di/Do
2440  Kw=FNKw(Ta(I))
2450  GOTO 2660
2460  END IF
2470  IF I=0 THEN
2480  Cf=1.
2490  Kw=FNKw(Ta(I))
2500  Hi=Kw/Di*Ci*Rei^Xn*Prw^.3333*Cf
2510  Dt=Q/(PI*Di*L*Hi)
2520  Cfc=(FNMu(Ta(I))/FNMu(Ta(I)+Dt))^.14
2530  IF ABS((Cfc-Cf)/Cfc)>.01 THEN
2540  Cf=(Cf+Cfc)/2.
2550  GOTO 2500
2560  END IF
2570  Ho=1./(Unr-Rw-Rfi*Do/Di-Do/(Di*Hi))
2580  Hir=1/Hi
2590  Qo=0
2600  ELSE
2610! COMPUTE 1/HI
2620  Hir=(Unr-1/Ho*(Q/Qo)^.3333-Rw-Rfi*Do/Di)*Di/Do
2630  Dt=Q/(PI*Di*L)*Hir
2640  Cf=(FNMu(Ta(I))/FNMu(Ta(I)+Dt))^.14
2650  END IF
2660  IF It=2 THEN
2670  Cf=1.
2680  Hic=Kw*Ci/Di*Rei^Xn*Prw^.3333*Cf
2690  Dt=Q/(PI*Di*L*Hic)
2700  Cfc=(FNMu(Ta(I))/FNMu(Ta(I)+Dt))^.14
2710  IF ABS((Cf-Cfc)/Cfc)>.01 THEN
2720  Cf=(Cf+Cfc)*.5
2730  GOTO 2680
```

```
2740  END IF
2750  END IF
2760  Qp=Q/(PI*Do*L)
2770  X=X/Cf
2780  IF Jj=1 THEN
2790  OUTPUT @Filep;X,Hir
2800  Hirc=A+B*X
2810  Ssq=Ssq+(Hir-Hirc)^2
2820  END IF
2830  IF J=1 OR Jj=1 THEN PRINT USING "10X,Z.7D,3X,Z.7D,4X,DD.DD,2X,D.3DE,X,Z.5D
";X,Hir,Dt,Qp,Cf
2840! COMPUTE COEFFICIENTS FOR LEAST-SQUARES SCHEME
2850  Sx=Sx+X
2860  Sy=Sy+Hir
2870  Sxs=Sxs+X^2
2880  Sxy=Sxy+X*Hir
2890  NEXT I
2900  N=K+1
2910! COMPUTE THE SLOPE OF THE LEAST-SQUARES LINE
2920! DEVELOPED FOR THE WILSON PLOT
2930  B=(N*Sxy-Sy*Sx)/(N*Sxs-Sx*Sx)
2940  Cic=Di/(B*Kw)
2950  PRINT USING "10X,""Intermediate value of Sieder-Tate coefficient = "",Z.4D
";Cic
2960  IF ABS((Ci-Cic)/Cic)>.01 THEN
2970  Ci=Cic
2980  GOTO 2260
2990  END IF
3000  IF Jj=1 THEN
3010  PRINT USING "10X,""Sieder-Tate coefficient  =  "",Z.4D";Cic
3020  A=(Sy-B*Sx)/N
3030  PRINT USING "10X,""Estimated fouling factor = "",MZ.5DE,"" (m^2-K/W)""";A
3040  PRINT "           Least-squares line:"
3050  PRINT USING "13X,""Slope     =  "",Z.5D";B
3060  PRINT USING "13X,""Intercept = "",MZ.5DE";A
3070  ELSE
3080  Jj=1
3090  GOTO 2260
3100  END IF
3110  PRINT USING "10X,""Sum of squares = "",D.5DE";Ssq
3120  ASSIGN @File TO *
3130  ASSIGN @Filep TO *
3140  IF Im=1 THEN
3150  BEEP
3160  PRINT USING "10X,""NOTE: "",DD,"" Data runs were stored in file "",8A";M,D
_file$
3170  END IF
3180  PRINT USING "10X,""NOTE: "",DD,"" X-Y pairs were stored in file "",10A";K+
1,Dplot$
3190  BEEP
3200  INPUT "ARE YOU RUNNING ANOTHER SERIES (1=YES,0=NO)?",Go_on
3210  IF Go_on=1 THEN Series
3220  END
3230  DEF FNRhow(T)
3240!
3250! THIS FUNCTION COMPUTES THE DENSITY OF WATER
3260!
3270  Ro=1006.35724-T*(.774489-T*(2.262459E-2-T*3.03304E-4))
3280  RETURN Ro
3290  FNEND
```

```
3300  DEF FNPvsv(V)
3310!
3320! THIS FUNCTION CONVERTS THE PRESSURE TRANSDUCER
3330! READING FROM VOLTS TO PRESSURE IN MM HG
3340!
3350  Y=1.1103462+163.36413*V
3360  RETURN Y
3370  FNEND
3380  DEF FNTvsp(P)
3390!
3400! THIS FUNCTION COMPUTES THE SATURATION TEMPERATURE
3410! CORRESPONDING TO PRESSURE IN MM HG
3420!
3430  IF P<600 THEN
3440  T=31.8776158+P*(.235854929-P*(3.6613664E-4-P*2.41652372E-7))
3450  ELSE
3460  T=59.36562+P*(.07379467-P*(3.15662E-5-P*6.27246E-9))
3470  END IF
3480  RETURN T
3490  FNEND
3500  DEF FNTemp(T,I)
3510!
3520! THIS FUNCTION APPLIES THERMOCOUPLE CORRECTIONS
3530!
3540  DIM A(5),B(5)
3550  DATA 0.640533,0.573054,0.593101,0.57298,0.567384,0.569577
3560  DATA 11.8744,8.63163,9.39412,8.570246,8.299436,8.36677
3570  READ A(*),B(*)
3580  T=T-(A(I)-B(I)*.001*T)
3590  RETURN T
3600  FNEND
3610  SUB Tvsv(T)
3620  COM /Cc/ C(7)
3630  Sum=0.
3640  FOR I=0 TO 7
3650  Sum=Sum+C(I)*T^I
3660  NEXT I
3670  T=Sum
3680  SUBEND
3690!
3700! THIS FUNCTION COMPUTES PRANDTL NUMBER FOR WATER
3710!
3720  DEF FNPrw(T)
3730  Pr=10^(1.09976605-T*(.013749326-T*(3.968875E-5+3.45026E-7*T)))
3740  RETURN Pr
3750  FNEND
3760  DEF FNMu(T)
3770!
3780! THIS FUNCTION COMPUTES THE VISCOSITY OF WATER
3790!
3800  Mu=1.5087546575E-3-T*(3.025732489E-5-T*(2.626439826E-7-T*8.18601937E-10))
3810  RETURN Mu
3820  FNEND
3830  DEF FNKw(T)
3840!
3850! THIS FUNCTION COMPUTES THE THERMAL CONDUCTIVITY OF WATER
3860!
3870  Kwa=.572183504477+1.52770121209E-3*T
3880  RETURN Kwa
3890  FNEND
```

C. PLOTTING PROGRAM

```
1000! FILE NAME: PLOT
1010  PRINTER IS 705
1020  BEEP
1030  INPUT "ENTER MINIMUM AND MAXIMUM X-VALUES",Xmin,Xmax
1040  BEEP
1050  INPUT "ENTER MINIMUM AND MAXIMUM Y-VALUES",Ymin,Ymax
1060  BEEP
1070  INPUT "ENTER STEP SIZE FOR X-AXIS",Xstep
1080  BEEP
1090  INPUT "ENTER STEP SIZE FOR Y-AXIS",Ystep
1100  BEEP
1110  PRINT "IN;SP1;IP 2300,1800,8300,6800;"
1120  PRINT "SC 0,100,0,100;TL 2,0;"
1130  Sfx=100/(Xmax-Xmin)
1140  Sfy=100/(Ymax-Ymin)
1150  PRINT "PU 0,0 PD"
1160  FOR Xa=Xmin TO Xmax STEP Xstep
1170  X=(Xa-Xmin)*Sfx
1180  PRINT "PA";X,",0; XT;"
1190  NEXT Xa
1200  PRINT "PA 100,0;PU;"
1210  PRINT "PU PA 0,0 PD"
1220  FOR Ya=Ymin TO Ymax STEP Ystep
1230  Y=(Ya-Ymin)*Sfy
1240  PRINT "PA 0,";Y,"YT"
1250  NEXT Ya
1260  PRINT "PA 0,100 TL 0 2"
1270  FOR Xa=Xmin TO Xmax STEP Xstep
1280  X=(Xa-Xmin)*Sfx
1290  PRINT "PA";X,",100; XT"
1300  NEXT Xa
1310  PRINT "PA 100,100 PU PA 100,0 PD"
1320  FOR Ya=Ymin TO Ymax STEP Ystep
1330  Y=(Ya-Ymin)*Sfy
1340  PRINT "PD PA 100,",Y,"YT"
1350  NEXT Ya
1360  PRINT "PA 100,100 PU"
1370  PRINT "PA 0,-2 SR 1.5,2"
1380  FOR Xa=Xmin TO Xmax STEP Xstep
1390  X=(Xa-Xmin)*Sfx
1400  PRINT "PA";X,",0;"
1410  PRINT "CP -2,-1;LB";Xa;""
1420  NEXT Xa
1430  PRINT "PU PA 0,0"
1440  FOR Ya=Ymin TO Ymax STEP Ystep
1450  Y=(Ya-Ymin)*Sfy
1460  PRINT "PA 0,";Y,""
1470  PRINT "CP -4,-.25;LB";Ya;""
1480  NEXT Ya
1490  BEEP
1500  INPUT "ENTER X-LABEL",Xlabel$
1510  BEEP
1520  INPUT "ENTER Y-LABEL",Ylabel$
1530  PRINT "SR 1.5,2;PU PA 50,-10 CP";-LEN(Xlabel$)/2;"0;LB";Xlabel$;""
1540  PRINT "PA -11,50 CP 0,";-LEN(Ylabel$)/2*5/6;"DI 0,1;LB";Ylabel$;""
1550  PRINT "CP 0,0 DI"
```

```
1560 Repeat:!
1570  BEEP
1580  INPUT "DO YOU WANT TO PLOT DATA FROM A FILE (1=YES,0=NO)?",Ok
1590  IF Ok=1 THEN
1600  BEEP
1610  INPUT "ENTER THE NAME OF THE DATA FILE",D_file$
1620  ASSIGN @File TO D_file$
1630  BEEP
1640  INPUT "ENTER THE BEGINNING RUN NUMBER",Md
1650  BEEP
1660  INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED",Npairs
1670  BEEP
1680  INPUT "SELECT A SYMBOL FOR THE PLOTTER (1=*,2=+,3=c,4=o,5= )",Sy
1690  PRINT "PU DI"
1700  IF Sy=1 THEN PRINT "SM*"
1710  IF Sy=2 THEN PRINT "SM+"
1720  IF Sy=3 THEN PRINT "SMc"
1730  IF Sy=4 THEN PRINT "SMo"
1740  IF Sy=5 THEN PRINT "SM "
1750  BEEP
1760  INPUT "SELECT MODE (1=HN/H1,2=HN(avg)/H1)",Jm
1770  IF Md>Npairs THEN
1780  FOR I=1 TO (Md-1)
1790  IF Jm=1 THEN ENTER @File:Xa,Ya,Yy
1800  IF Jm=2 THEN ENTER @File;Xa,Yy,Ya
1810  NEXT I
1820  END IF
1830  FOR I=1 TO Npairs
1840  IF Jm=1 THEN ENTER @File:Xa,Ya,Yy
1850  IF Jm=2 THEN ENTER @File:Xa,Yy,Ya
1860  X=(Xa-Xmin)*Sfx
1870  Y=(Ya-Ymin)*Sfy
1880  PRINT "PA",X,Y,""
1890  NEXT I
1900  BEEP
1910  ASSIGN @File TO *
1920  INPUT "DO YOU HAVE MORE DATA TO BE PLOTTED (1=YES,0=NO)?",Go_on
1930  IF Go_on=1 THEN Repeat
1940  END IF
1950  BEEP
1960  INPUT "DO YOU LIKE TO PLOT THE NUSSELT RELATION (1=YES,0=NO)?",Go_on
1970  PRINT "PU:SM"
1980  IF Go_on=1 THEN
1990  FOR Xa=Xmin TO Xmax STEP Xstep/50
2000  X=(Xa-Xmin)*Sfx
2010  IF Jm=1 AND Xa>Xmin THEN Ya=Xa^.75-(Xa-1)^.75
2020  IF Jm=2 AND Xa>Xmin THEN Ya=Xa^(-.25)
2030  IF Xa=Xmin THEN Ya=1
2040  Y=(Ya-Ymin)*Sfy
2050  PRINT "PA",X,Y,"PD"
2060  NEXT Xa
2070  BEEP
2080  PRINT "PU"
2090  INPUT "MOVE THE PEN TO LABEL THE NUSSELT LINE",Ok
2100  PRINT "LBNusselt"
2110  END IF
2120  BEEP
2130  INPUT "DO YOU LIKE TO PLOT KERN RELATIONSHIP?",Yes
2140  IF Yes=1 THEN
2150  FOR Xa=Xmin TO Xmax STEP Xstep/20
```

```
2160  Ya=Xa^(-1/6)
2170  X=(Xa-Xmin)*Sfx
2180  Y=(Ya-Ymin)*Sfy
2190  PRINT "PA",X,Y,"PD"
2200  NEXT Xa
2210  PRINT "PU"
2220  BEEP
2230  INPUT "MOVE THE PEN TO LABEL KERN RELATIONSHIP",Ok
2240  PRINT "LBKern:PU"
2250  END IF
2260  PRINT "PU PA 0,0"
2270  BEEP
2280  INPUT "OK TO PLOT CURVE FOR NEW DESIGN (1=OK,0=NO)?",Ok
2290  IF Ok=1 THEN
2300  FOR Xa=Xmin TO Xmax STEP Xstep/2
2310  IF Xa=1 THEN Ya=1
2320  IF Xa>1 THEN Ya=((Xa*5)^.75-(Xa*5-1)^.75+5)/6
2330  X=(Xa-Xmin)*Sfx
2340  Y=(Ya-Ymin)*Sfy
2350  PRINT "PA",X,Y,"PD"
2360  NEXT Xa
2370  PRINT "PU"
2380  END IF
2390  BEEP
2400  INPUT "DO YOU LIKE TO PLOT EISSENBERG RELATION?",Go_on
2410  IF Go_on=1 THEN
2420  FOR Xa=Xmin TO Xmax STEP Xstep/10
2430  Ya=.6+.42*Xa^(-.25)
2440  X=(Xa-Xmin)*Sfx
2450  Y=(Ya-Ymin)*Sfy
2460  PRINT "PA",X,Y,"PD"
2470  NEXT Xa
2480  PRINT "PU"
2490  BEEP
2500  INPUT "MOVE THE PEN TO LABEL THE EISSENBERG LINE",Ok
2510  PRINT "LBEissenberg:PU"
2520  END IF
2530  PRINT "PU"
2540  BEEP
2550  INPUT "DO YOU LIKE TO PLOT THE EXPTL CURVE (1=Y,0=NO)?",Go_on
2560  IF Go_on=1 THEN
2570  BEEP
2580  INPUT "ENTER THE EXPONENT",Ex
2590  FOR Xa=Xmin TO Xmax STEP Xstep/10
2600  Ya=Xa^(-Ex)
2610  X=(Xa-Xmin)*Sfx
2620  Y=(Ya-Ymin)*Sfy
2630  PRINT "PA",X,Y,"PD"
2640  NEXT Xa
2650  PRINT "PU"
2660  BEEP
2670  INPUT "MOVE THE PEN TO LABEL THE EXPTL CURVE",Ok
2680  PRINT "LBHN(avg)/H1=N"
2690  PRINT "PR 1,1"
2700  PRINT "LB";-Ex;""
2710  END IF
2720  BEEP
2730  INPUT "DO YOU LIKE TO DRAW A STRAIGHT LINE?",Go_on
2740  IF Go_on=1 THEN
2750  BEEP
2760  INPUT "ENTER THE SLOPE",Sl
```

```
2770  BEEP
2780  INPUT "ENTER THE INTERCEPT",Ac
2790  FOR Xa=Xmin TO Xmax STEP (Xmax-Xmin)
2800  Ya=Ac+Si*Xa
2810  Y=(Ya-Ymin)*Sfy
2820  X=(Xa-Xmin)*Sfx
2830  IF Y<0 THEN
2840  Xam=(Ymin-Ac)/Sl
2850  X=(Xam-Xmin)*Sfx
2860  Y=0
2870  END IF
2880  IF Y>100 THEN
2890  Xam=(Ymax-Ac)/Sl
2900  X=(Xam-Xmin)*Sfx
2910  Y=100
2920  END IF
2930  PRINT "PA",X,Y,"PD"
2940  NEXT Xa
2950  END IF
2960  PRINT "PU SP0"
2970  END
```

LIST OF REFERENCES

1. Standards for Steam Surface Condensers, 6th ed., Heat Exchange Institute, 1970.

2. Standards of Tubular Exchange Manufacturers Association, 4th ed., Tubular Exchanger Manufacturers Association, Inc., 1959.

3. Search, H.T., A Feasibility Study of Heat Transfer Improvement in Marine Steam Condensers, MSME, Naval Postgraduate School, Monterey, CA, January 1977.

4. Webb, R.L., "The Use of Enhanced Surface Geometries in Condensers: An Overview," Power Condenser Heat Transfer Technology, Marto, P.J. and Nunn, R.H. (Eds), Hemisphere Publishing Corp., New York, 1980, pp. 287-318.

5. Beck, A.C., A Test Facility to Measure Heat Transfer Performance of Advanced Condenser Tubes, MSME, Naval Postgraduate School, Monterey, CA, March 1978.

6. Pence, D.T., An Experimental Study of Steam Condensation on a Single Horizontal Tube, MSME, Naval Postgraduate School, Monterey, CA, March 1978.

7. Reilly, D.J., An Experimental Investigation on Enhanced Heat Transfer on Horizontal Condenser Tubes, MSME, Naval Postgraduate School, Monterey, CA, March 1978.

8. Fenner, J.H., An Experimental Comparison of Enhanced Heat Transfer Condenser Tubing, MSME, Naval Postgraduate School, Monterey, CA, September 1978.

9. Ciftici, H., An Experimental Study of Filmwise Condensation on Horizontal Enhanced Condenser Tubing, MSME, Naval Postgraduate School, Monterey, CA, December 1979.

10. Marto, P.J., Reilly, D.J., Fenner, J.H., "An Experimental Comparison of Enhanced Heat Transfer Tubing," Advances in Enhanced Heat Transfer, Proceeding of the 18th National Conference, American Society of Mechanical Engineers, New York, NY, 1979.

11. Eissenberg, D.M., An Investigation of the Variables Affecting Steam Condensation on the Outside of a Horizontal Tube Bundle, Ph.D Thesis. University of Tennessee. Knoxville. TN, December 1972.

12. Noftz, P.J., Effects of Condensate Inundation and Vapor Velocity on Heat Transfer in a Condenser Tube Bundle, MSME, Naval Postgraduate School, Monterey, CA. June 1982.

13. Morrison, R.H., A Test Condenser to Measure Condensate Inundation Effects in a Tube Bundle, MSME. Naval Postgraduate School, Monterey, CA, March 1981.

14. Thomas, A., Lorenz, J.J., Hillis. D.A., Young, D.T. and Sather. N.F., "Performance Tests of the 1 Mwt Shell and Tube Exchangers for OTEC", Proc. 6th OTEC Conf., Paper 3c. June 19-22, 1979.

15. Cunningham. J., "The Effect of Noncondensable Gases on Enhanced Surface Condensation." Power Condenser Heat Transfer Technology, Marto, P.J. and Nunn, R.H. (Eds), Hemisphere Publishing Corp. New York, 1980, pp. 353-365.

16. Newson. I.H., "Enhanced Heat Transfer Condenser Tubing for Advanced Multistage Flash Distillation Plants", 5th Int. Symp. on Fresh Water from Sea, Vol. 2, pp. 107-115, 1976.

17. Eissenberg, D. and Bugue. Do. "Tests on an Enhanced Horizontal Tube Condenser Under Conditions of Horizontal Steam Cross Flow," 4th Int. Heat Transfer Conf., Versailles. H.E. 2, I, September 1976.

18. Nobbs, D.W., The Effect of Downward Vapour Velocity and Inundation on the Condensate Rates on Horizontal Tubes and Tube Banks, Ph.D Thesis, University of Bristol. Bristol, England. April 1975.

19. Jakob, Max, Heat Transfer, Vol. 1, J. Wiley & Sons, Inc., 1948, pp. 667-673.

20. Kern, D.Q., "Mathematical Development of Loading in Horizontal Condensers." AICHE Journal, Vol. 4, 1958, pp. 157-160.

21. Chen. M.M., "An Analytical Study of Laminar Film Condensation: Part 2 - Single and Multiple Horizontal Tubes." Journal of Heat Transfer, February 1965.

22. Berman. L.D., "Heat Transfer with Steam Condensation on a Bundle of Horizontal Tubes," Thermal Engineering, Vol. 28, 1981, pp. 218-224.

23. Naval Postgraduate School Report NPS69-82-005, Computer Rating of Enhanced Marine Condensers, by R.H. Nunn and P.J. Marto, p. 13, August 1982.

24. Berman, L.D. and Tumanov. Y.A., "Investigation of Heat Transfer in the Condensation of Moving Steam on a Horizontal Tube," Teploenergitika, Vol. 9, 1962, pp. 77-83.

25. Eissenberg, D.M., "Combined Effects of Vapor Shear and Inundation in Tube Bundles." Power Condenser Heat Transfer Technology, Marto. P.J. and Nunn, R.H. (Eds), Hemisphere Publishing Corp, New York, 1980, pp. 225-227.

26. Metals Handbook, 8th Ed., Vol 2, p. 665, American Society for Metals, 1964.

27. Holman. J.P., Heat Transfer, 4th Ed., McGraw-Hill, 1976.

28. Wilson, E.E., A Basis for Rational Design of Heat Transfer Apparatus, paper presented at the Spring Meeting of the Society of Mechanical Engineers, Buffalo. NY, June 1935.

29. Wanniarachchi. S.A., "Modified Wilson Plot Technique," private communication. January-June 1983.

30. Incropera, P.F. and DeWitt. P.D., Fundamentals of Heat Transfer, John Wiley & Sons.

31. Kline. S.J. and McClintock, F.A., Describing Uncertainties in Single Experiments. Mech. Engin., Vol. 74, pp. 3-8, January 1953.

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